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Experimental stress analysis of bolts in a flange coupling

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EXPERIMENTAL STRESS ANALYSIS OF BOLTS IN A

FLANGE COUPLING

BY

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A

THESIS

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ABSTRACT

This thesis deals with the strength of a bolted joint which is subjected to tensile and bending loads. The purpose of this investigation was to derive expressions for the stresses developed in the bolts and the load producing separation of the joint.

A laboratory study of a flange coupling was made and the behaviour of flange bolts under different clamping forces was studied in tension and bending tests. The strain in the bolt was measured by mounting the strain gages on the shank of the bolt. From this, a graph of applied load versus strain in the bolt was plotted. The load producing separation of the flanges was determined from the graph and was used in deriving the formula for the prediction of load which will loosen the joint.

The following factors will influence the strength of the flange coupling:-

(i) Initial clamping force on the bolts.

(ii) Thickness of the flange.

(iii) Number of bolts.

(iv) Diameter of the bolts.

(v) Diameter of the flange.

(vi) Material and thickness of the gasket.

The previous investigation of Mr. Vijay Rastogi*(1) deals with the first three factors and it is hoped that the reader is familiar

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*All references are in bibliography.

with his thesis. The author has made an attempt to investigate the remaining factors which will adequately supplement the analysis of a flange coupling.

ACKNOWLEDGEMENT

The author expresses his sincere thanks to Professor J. A. Jones for his invaluable guidance and interest in this investigation.

The author would also like to thank Dr. A. J. Miles, Chairman of the Mechanical Engineering Department, and Professor R. F. Davidson, Chairman of the Mechanics Department, for making the facilities of their departments available for use in this investigation. Their cooperation and encouragement has made this work possible.

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SYMBOLS AND UNITS

F_{bi} .... Initial clamping force, (lb.)

F_{fi} .... Initial force in the flange, (lb.)

δ_{bi} .... Initial deformation of bolt, (in.)

δ_{fi} .... Initial deformation of flange, (in.)

K_b .... Stiffness of bolt, (lb./in.)

K_f .... Stiffness of flange, (lb./in.)

F_a .... Applied load, (lb.)

δ_b .... Additional deformation of bolt, (in.)

F_b .... Total force in the bolts, (lb.)

F_f .... Total force in the flange, (lb.)

E_b .... Modulus of elasticity of bolt, (lb./in.^{2})

E_f .... Modulus of elasticity of flange, (lb./in.^{2})

A_b .... Area of the bolts, (in.^{2})

A_f .... Area of the flange, (in.^{2})

ϵ_{bi} .... Initial strain in the bolt, (in./in.)

ΔS .... Additional stress in the bolt, (lb./in.^{2})

S_b .... Stress in the bolt, (lb./in.^{2})

F_{al} .... Load required for separation, (lb.)

L_b .... Length of the bolt (in.)

L_f .... Thickness of the flange, (in.)

d .... Diameter of the bolt, (in.)

D_b .... Diameter of bolt circle, (in.)

D_f .... Diameter of flange, (in.)

M .... Applied bending moment, (lb. in.)

K_1, K_2, K_3 .... Constants determined by experiment.

I. INTRODUCTION

The design of an assembly of two or more metal parts fastened together with bolts, appears to be a rather simple problem in stress analysis, however, little fundamental information is available on which to base a rational design procedure. Some designers assume that the compression members of the assembly are rigid and do not deform when the bolts are tightened. In fact, no metal sections are incompressible and actual stress in the bolt is changed when the external load is applied irrespective of the clamping force. Hence, a better and more useful approach to the design of a bolted joint is necessary.

In this thesis, a flange coupling is studied under different initial tightening torques. An investigation has been done by Mr. Vijay Rastogi in this specific field. The author has made an attempt to supplement the previous work. The author has studied the influence of the following factors on the strength of the flange coupling under tensile and bending loads:

(i) The amount of the clamping force in the bolts.

(ii) Diameter of the bolts.

(iii) Diameter of the flange.

(iv) Material and thickness of the gaskets.

An experiment was conducted and the results of the experiment were used to determine the stresses and strength of a bolted joint. Major considerations were placed on tensile and bending loads. So, results are only applicable to these situations.

Careful analysis should be made for a joint which is subjected to repeated loading. Various investigations have been done in the past to determine the fatigue strength of a bolted joint. (2,3). Hence, reference should be made according to the conditions of the problem.

II. THEORETICAL ANALYSIS OF FLANGE COUPLING IN TENSION

The following assumptions are made in the theoretical analysis of a flange coupling in tension.

1. The bolt and the flanges are perfectly elastic. So, deformation is proportional to the applied load. (Hooke's Law.)
2. The bolt elongates a definite amount due to an applied load.
3. The whole flange experiences a uniform compressive stress due to tightening of the bolts.
4. There are no stress concentrations around the bolts.
5. The joint behaves as a single unit, i.e., the elongation of the bolt is equal to the elongation of the flange due to external load before separation occurs.
6. Change in cross-sectional area of the bolt due to it's increase in length is negligible.

These assumptions are quite legitimate and they are necessary to limit the scope of experimental investigation.

Figure 1 shows the flange coupling when subjected to pure tension. Due to tightening of the bolts, the flanges are compressed and the bolts are elongated. This can be schematically represented as a system which consists of two springs as shown in Figure 2. The inner spring represents the bolts and it is extended, the outer spring represents the flanges and it is compressed. This analogy is valid until the flanges separate.

Due to initial clamping, the bolt elongates elastically by an amount δ_{bi}, while the flanges are compressed by an amount δ_{fi}. But, the

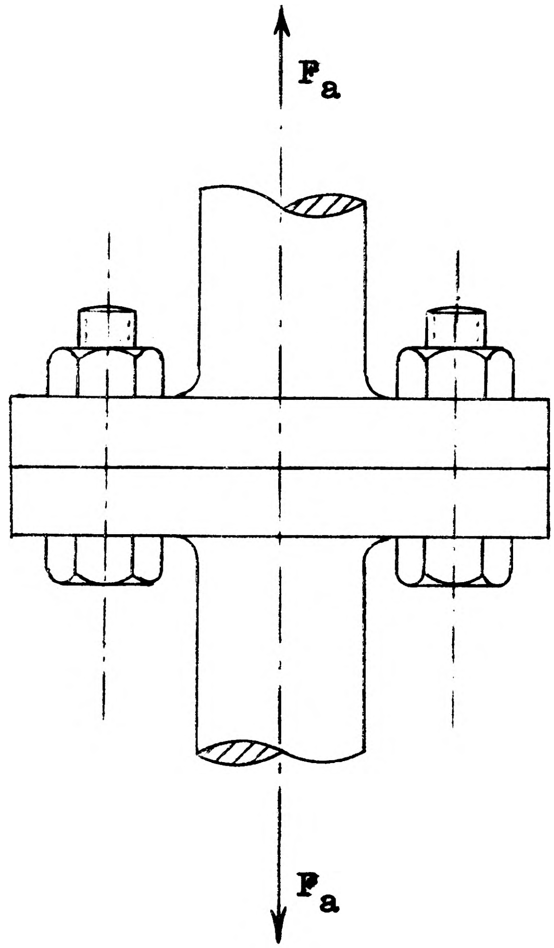

Figure 1. Flange Coupling in Tension

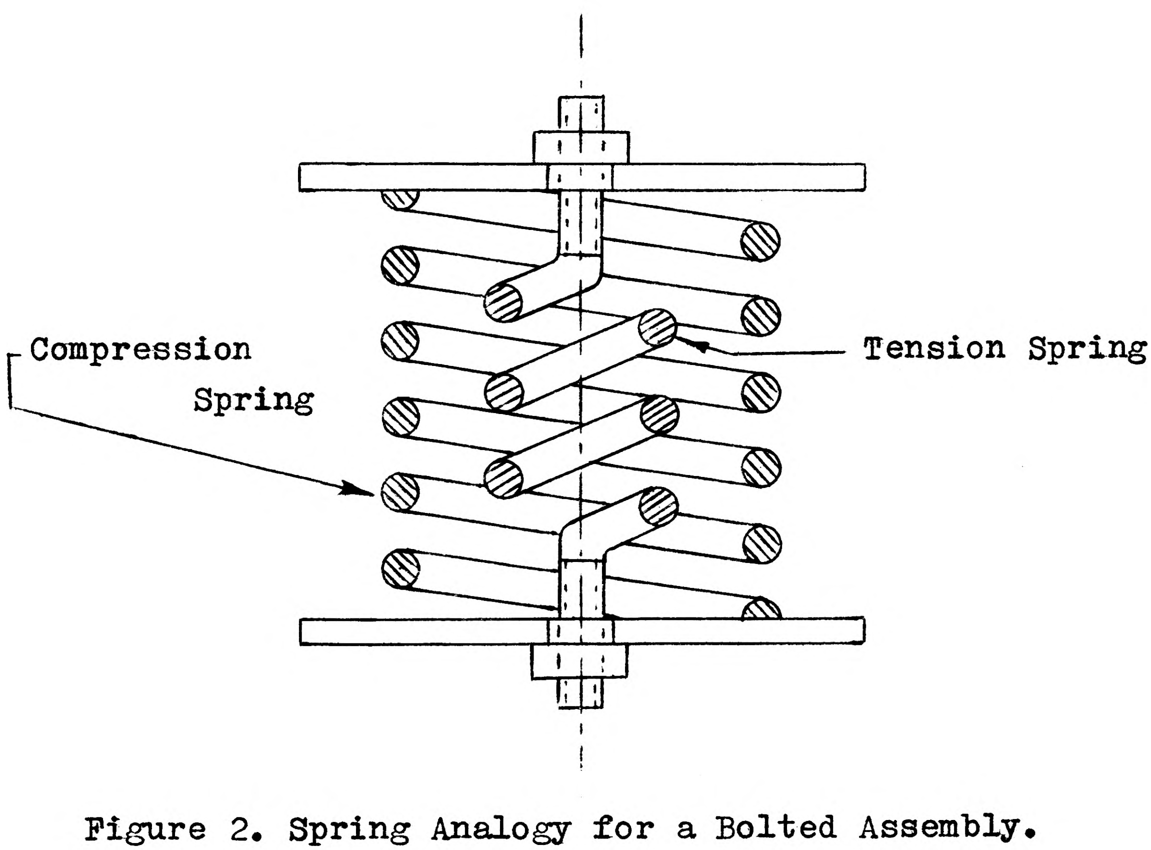

Figure 2. Spring Analogy for a Bolted Assembly.

tensile force in the bolts is equal to the force of compression in the flanges. Figure 3 shows the Load - Deformation relationship for a bolted assembly.

From the Figure 3,

1. Initial clamping force in the bolt = Initial compressive force in the flange

2. $F_{bi} = F_{fi}$

Or, $(\delta_{bi} \cdot K_b) = (\delta_{fi} \cdot K_f)$

By applying the external force F_a, the bolt experiences an additional elongation δ_b, and the compression of the flanges decreases by the same amount. For the condition of equilibrium:-

3. $F_b = F_a + F_f$

Or, $F_a = F_b - F_f$

4. But, $F_b = F_{bi} + (\delta_b \cdot K_b)$

5. $F_f = F_{fi} - (\delta_f \cdot K_f)$

Substituting these values in equation 3.

6. $F_a = F_{bi} + (\delta_b \cdot K_b) - F_{fi} + (\delta_f \cdot K_f)$

7. But, $F_{bi} = F_{fi}$ and $\delta_b = \delta_f$

8. Hence, $F_a = \delta_b (K_b + K_f)$

9. Now, $K_b = \dfrac{E_b \cdot A_b}{L_b}$, $K_f = \dfrac{E_f \cdot A_f}{L_f}$

10. So, $F_a = \delta_b \left(\dfrac{E_b \cdot A_b}{L_b} + \dfrac{E_f \cdot A_f}{L_f}\right)$

Taking $E_b = E_f$ and $L_b = L_f$

11. $F_a = \delta_b \cdot \dfrac{E_b}{L_b} (A_b + A_f)$

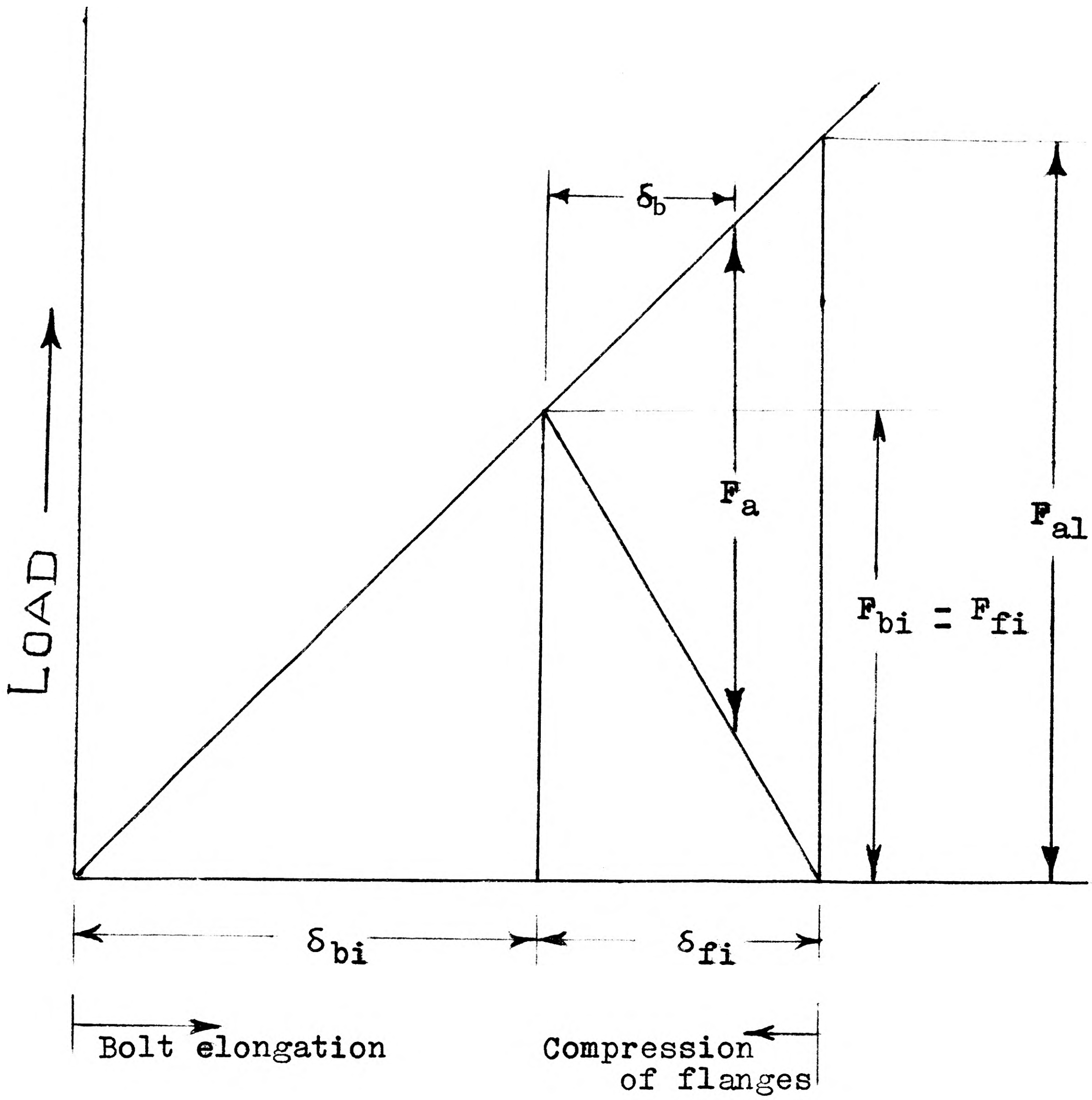

Figure 3. Load - Deformation Relationship for Bolted Assembly.

12. But, $\Delta S = \dfrac{\delta_b \cdot E_b}{L_b}$

13. Hence, $F_a = \Delta S \; (A_b + A_f)$

Since the entire flange is not effective in resisting the external load,

14. $F_a = \Delta S \; (A_b + K_1 \; A_f)$

Where K_1 is a constant which will be determined experimentally.

15. $S_b = \dfrac{F_{bi}}{A_b} + \dfrac{F_a}{(A_b + K_1 \; A_f)}$

Figure 3 shows that the flanges will be separated when the additional elongation of the bolt is equal to the initial compression of the flange.

16. Hence, when separation occurs,

$$\delta_b = \delta_{fi}$$

17. $F_{bi} = (\delta_{bi} \cdot K_b) = F_{fi} = (\delta_{fi} \cdot K_f)$

18. So, $\delta_{fi} = \delta_{bi} \dfrac{K_b}{K_f} = \delta_{bi} \dfrac{A_b \cdot E_b}{A_f \cdot E_f} = \delta_{bi} \dfrac{A_b}{K_2 \; A_f} = \delta_b$

Where K_2 is a ratio of the effective area in compression to the actual flange area.

Substituting equation 18 in equation 11,

19. $F_{a1} = \delta_{bi} \cdot \dfrac{A_b \cdot E_b}{K_2 \; A_f \; L_b} \; (A_b + K_1 A_f)$

But, $\dfrac{\delta_{bi}}{L_b} = \epsilon_{bi}$

20. Finally,

$$F_{a1} = \frac{\epsilon_{bi} \cdot A_b \cdot E_b \; (A_b + K_1 \; A_f)}{K_2 \; A_f}$$

The values of K_1, K_2 will be determined experimentally.

Then equation 20 will give the amount of load which will loosen the joint for a particular value of initial strain.

III. THEORETICAL ANALYSIS OF A FLANGE COUPLING IN BENDING

The following assumptions are made in the theoretical analysis of a flange coupling in bending.

1. The bolt and the members of the assembly are perfectly elastic and all parts conform with Hooke's Law, i.e., the joint will develop only elastic deformation.
2. The bolt elongates a definite amount due to an applied bending moment, when the bolt is on the tension side of the neutral axis.
3. The whole flange experiences a uniform compressive stress due to the initial tightening of the bolts.
4. There is no stress concentration around the bolts.
5. Change in cross-sectional area of the bolts due to their increase in length is negligible.

These assumptions are quite legitimate and they are necessary to limit the scope of experimental investigation.

Figure 4 shows the flange coupling in bending. The initial clamping force in the instrumented bolt is given by

21. $F_c = \epsilon_{bi} \cdot E_b \cdot A_b$

By applying a bending moment to the joint, the lower portion of the flange is subjected to tensile stresses. These stresses will produce a tensile force. This force is given by

22. $F_t = \frac{M \cdot C}{I} \quad (A_f \cdot K_3)$

Where, M = Bending moment at the center

C = Distance from the neutral axis = $\frac{D_b}{2}$

I = Moment of Inertia of the flange = $\frac{\pi}{64} \cdot D^4_f$

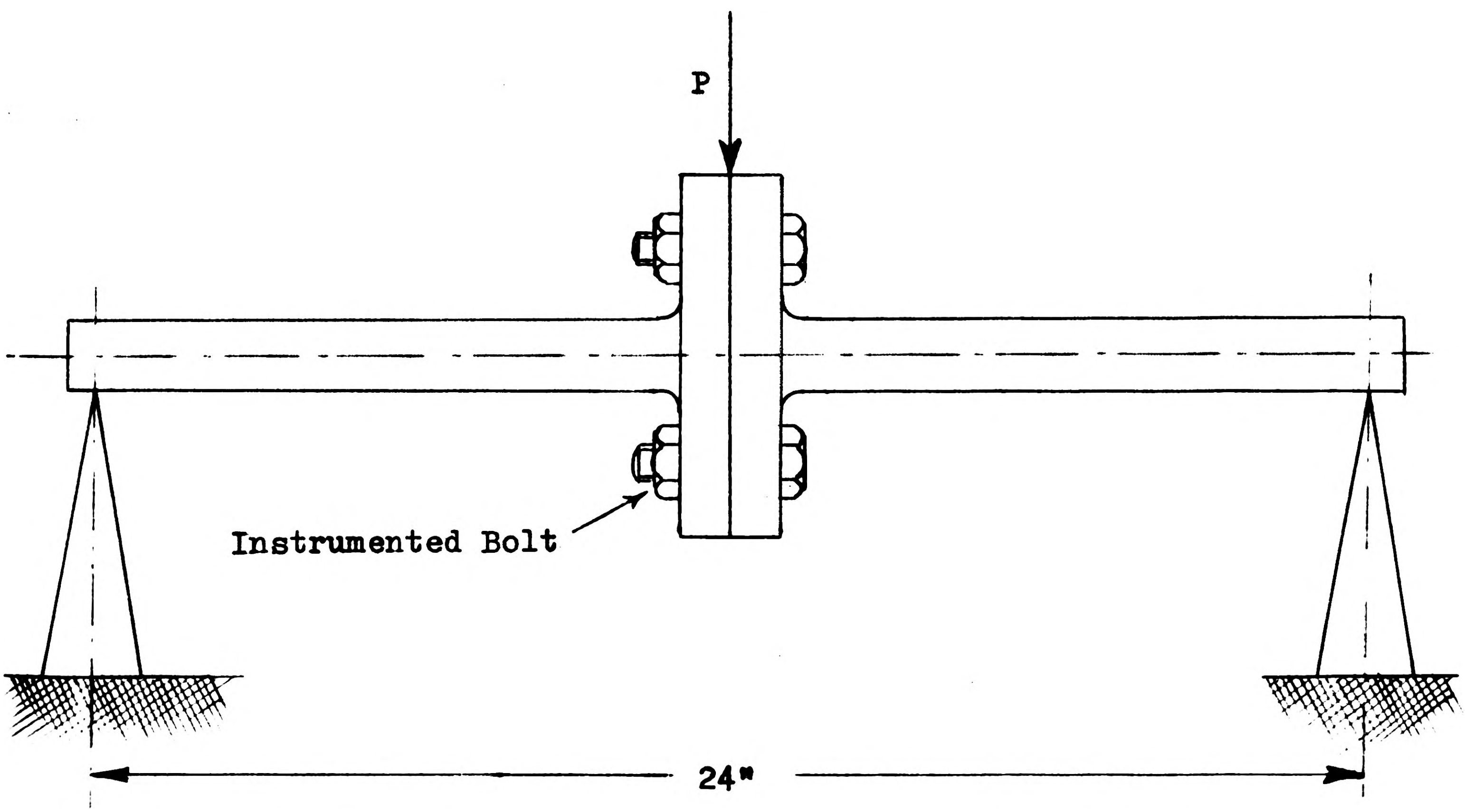

Figure 4. Diagram of Flange Coupling in Bending

A_f = Area of the flange = $\frac{\pi}{4} \cdot D^2_f$

K_3 = Constant which is the ratio of the effective flange area in tension to the total flange area.

The separation of the flanges will occur when the tensile force F_t, is equal to, or greater than the initial clamping force F_c. Hence, at the point of separation:-

23. $$\frac{M \cdot C}{I} \cdot (A_f \cdot K_3) = (\epsilon_{bi} \cdot E_b \cdot A_b)$$

Substituting the values for I, C, and A_f

24. $$\frac{M \cdot \frac{D_b}{2}}{\frac{\pi}{64} \cdot D^4_f} \left(\frac{\pi}{4} D^2_f \cdot K_3 \right) = (\epsilon_{bi} \cdot E_b \cdot A_b)$$

Simplifying,

25. $$M = \frac{\epsilon_{bi} \cdot E_b \cdot A_b \cdot D^2_f}{8 \cdot D_b \cdot K_3}$$

The constant K_3 will be determined experimentally. Then equation 25 will give the value of bending moment which will loosen the joint for a given initial strain.

IV. DISCUSSION

In this investigation, an experiment was designed to determine the influence of various factors on the strength of a bolted joint. A typical study of flange coupling with three bolts was made under tension and bending loads. The following will give the detailed information on the equipment used, procedure followed, interpretation of data, and results.

FLANGE COUPLINGS:-

The size of the flange coupling was limited by the testing machine available. The Universal Testing Machine which was used in tension and bending tests, can hold bars up to 1.5 inches diameter. Hence, a 1.25" diameter bar was used. Figure 5 shows the dimensions for the Standard flange coupling of 1.25 inches diameter bar.

BOLT INSTRUMENTATION:-

Three bolts of 1/2" diameter were used as fasteners. One of the bolts was instrumented by mounting two SR-4 strain gages diametrically opposite to each other in the longitudinal direction, on the shank of the bolt. The specifications for the SR-4 strain gages are as follows:-

Type:- A-18

Resistance:- 120.0 $\pm$ 0.3 ohms.

Gage Factor:- 1.79 $\pm$ 2.0%

Company:- Baldwin-Lima-Hamilton Corporation.

The two strain gages were connected in series, so that any deformation due to bending of the bolts could be compensated.

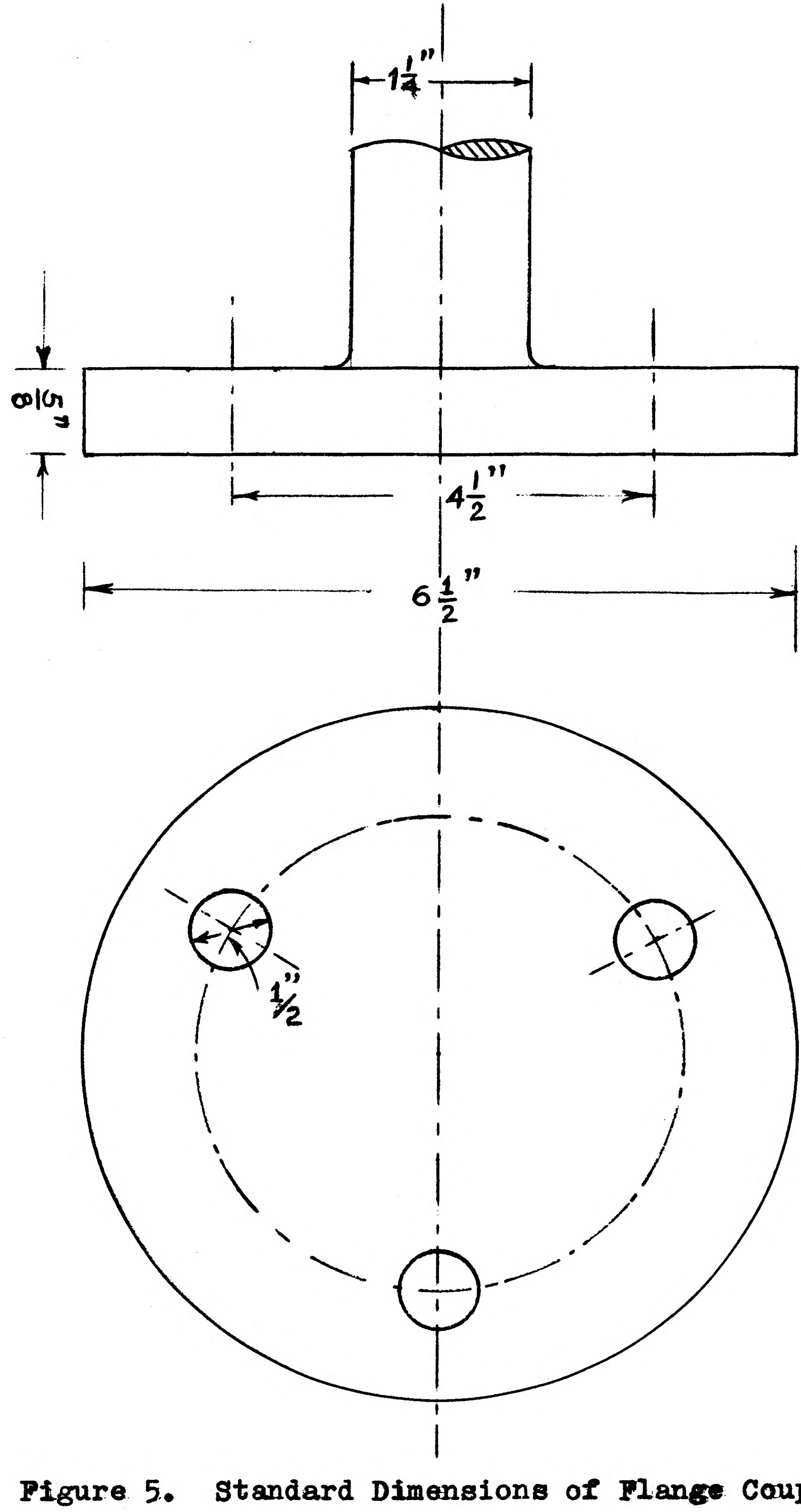

Figure 5. Standard Dimensions of Flange Coupling

In order to read the strain directly in microinches per inch, two strain gages of the same type were used for the compensating gage. Figure 6 shows the instrumented bolt.

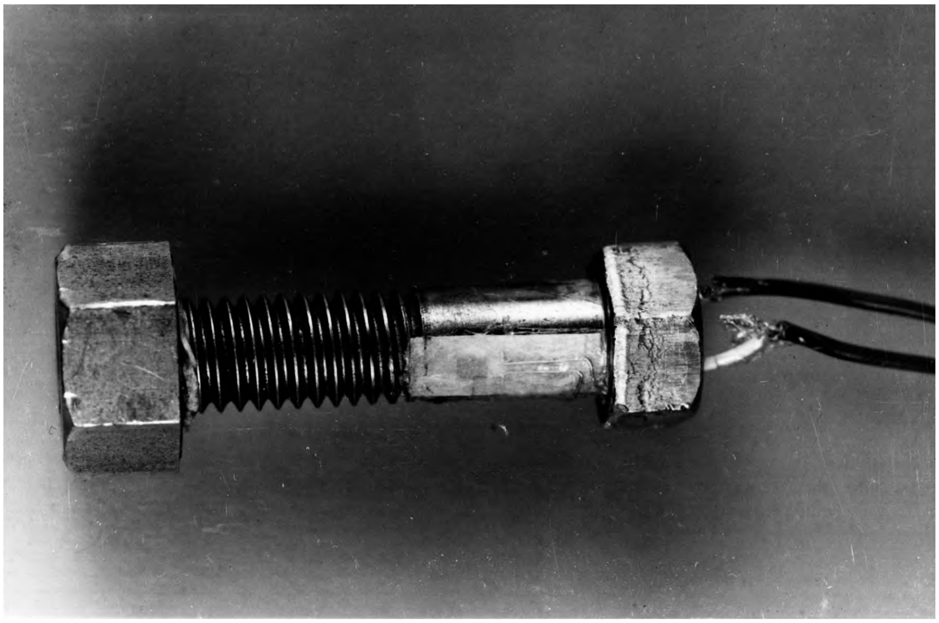

Figure 6. The Instrumented Bolt.

A. Testing of Flange Coupling With Varying Diameter of Bolts

The purpose of this experiment was to determine the influence of bolt diameter on the strength of the joint. The bolt diameter was varied from 3/8" to 3/4" in steps of 1/8". Each time, the flange coupling was tested in tension and bending. The following is the description of the procedure employed throughout the experiment.

1. TENSION TEST

In this test, the flange coupling was subjected to tensile load by the testing machine. Figure 7 shows the flange coupling in tension. The Universal Testing Machine, which is shown in the photograph, has the following specifications:-

Capacity:- 60,000 lbs.

Load Speed:- 0.025" to 2.0" per minute.

Company:- American Machines & Metal Inc.

RIEHLE Testing Machines Division

The initial clamping force was applied by a calibrated torque wrench. Table No. 1 shows the data obtained in a tension test of 3/8" diameter bolts. Similar data was obtained for 1/2", 5/8", and 3/4" diameter bolts. From this data, the graphs of applied load versus strain were plotted. The load causing separation of the flanges was obtained when a sudden change in slope was observed, as shown in Figure 8. All graphs were corrected to the same origin. Sample calculations for the 3/8" diameter bolt are given on page 29.

Figure 7. The Flange Coupling in Tension Test.

TABLE I

SAMPLE DATA FOR TENSION TEST

Flange diameter:-	6.5"
Flange thickness:-	5/8"
Bolt circle diameter:-	4.5"
Diameter of the bolt:-	3/8"
Number of bolts:-	3
Initial meter reading:-	11,468

INITIAL TIGHTENING TORQUE:- 100 in. lb.

Load (lbs.)	Meter Reading	Strain x 10^6 (in/in.)
0	11,600	0
500	11,614	14
1,000	11,630	30
1,500	11,660	60
2,000	11,690	90
2,500	11,730	130
3,000	11,754	154
3,500	11,790	190
4,000	11,830	230

TABLE I. Continued.

INITIAL TIGHTENING TORQUE:- 200 in. lb.

Load (lbs.)	Meter Reading	Strain x 10^6(in./in.)
0	11,802	0
1,000	11,817	15
1,500	11,832	30
2,000	11,842	40
2,500	11,872	70
3,000	11,902	100
3,500	11,922	120
4,000	11,961	159
4,500	11,982	180
5,000	12,022	220

TABLE I Continued.

INITIAL TIGHTENING TORQUE:- 300 in. lb.

Load (lbs.)	Meter Reading	Strain x 10^6(in./in.)
0	12,027	0
1,500	12,047	20
2,000	12,057	30
2,500	12,067	40
3,000	12,087	60
3,500	12,097	70
4,000	12,127	100
4,500	12,157	130
5,000	12,187	160
5,500	12,207	180
6,000	12,247	220

INITIAL TIGHTENING TORQUE:- 400 in. lb.

Load (lbs.)	Meter Reading	Strain x 10^6(in./in.)
0	12,383	0
2,000	12,403	20
3,000	12,423	40
4,000	12,453	70
5,000	12,473	90
6,000	12,513	130
7,000	12,553	170
8,000	12,603	220

<u>Sample Calculations For Tension Tests</u>

Using 3/8" diameter bolt,

From Figure 8, at a load of 3,000 lbs., the strain in the bolts, before separation of the flanges, is 73×10^{-6} in./in.

So, $\Delta S = \epsilon E = 73 \times 10^{-6} \times 30 \times 10^{6} = 2{,}190$ lb./in.^{2}

Area of the flange $= \frac{\pi}{4} D_f^2 = \frac{\pi}{4} \times (6.5)^2 = 33.2$ in.^{2}

Area of the bolts $= 3 \times \frac{\pi}{4} \times (3/8)^2 = 0.3318$ in.^{2}

From equation 14,

$$F_a = \Delta S \, (A_b + K_1 A_f)$$

$$3{,}000 = 2{,}190 \, (0.3318 + K_1 \times 33.2)$$

So, $\underline{K_1 = 0.0312}$

<u>Load Causing Separation Of The Flanges</u>:-

For initial tightening torque of 300 in. lb., the initial strain in the bolts is 559×10^{-6} in./in. The load causing separation is given by equation 20,

$$F_{a1} = \frac{\epsilon_{bi} \cdot A_b \cdot E_b \cdot (A_b + K_1 A_f)}{K_2 \cdot A_f}$$

From the Figure 8, at 300 in. lb. torque, the load producing separation of the flanges is 3,400 lbs.

Hence,

$$3{,}400 = \frac{559 \times 10^{-6} \times 30 \times 10^{6} \; (0.3318 + 0.0312 \times 33.2) \times 0.3318}{K_2 \times 33.2}$$

So, $\underline{K_2 = 0.0674}$

Effective Area $= K_2 A_f = 0.0674 \times 33.2 = \underline{2.239 \text{ in.}^2}$

Now,

$$F_{a1} = \frac{\epsilon_{bi} \times 30 \times 10^6 \times 0.3318 \times (0.3318 + 0.0312 \times 33.2)}{2.239}$$

$$\underline{F_{a1} = 6.08 \times 10^6 \times \epsilon_{bi}}$$

The initial tension in the bolt is given by,

$$F_{bi} = \epsilon_{bi} \times E_b \times A_b$$

$$= \epsilon_{bi} \times 30 \times 10^6 \times \frac{\pi}{4} \times (3/8)^2$$

$$\underline{F_{bi} = 3.318 \times 10^6 \times \epsilon_{bi}}$$

Table III shows the predicted and actual values of the load causing separation of the flanges for different bolt sizes at different initial tension in the bolts.

TABLE II

LEGEND FOR TENSION TEST GRAPHS

	Initial Tightening Torque
—⊙—⊙—⊙—	100 in. lb.
—×—×—×—	200 in. lb.
—△—△—△—	300 in. lb.
—▽—▽—▽—	400 in. lb.
—⊡—⊡—⊡—	500 in. lb.

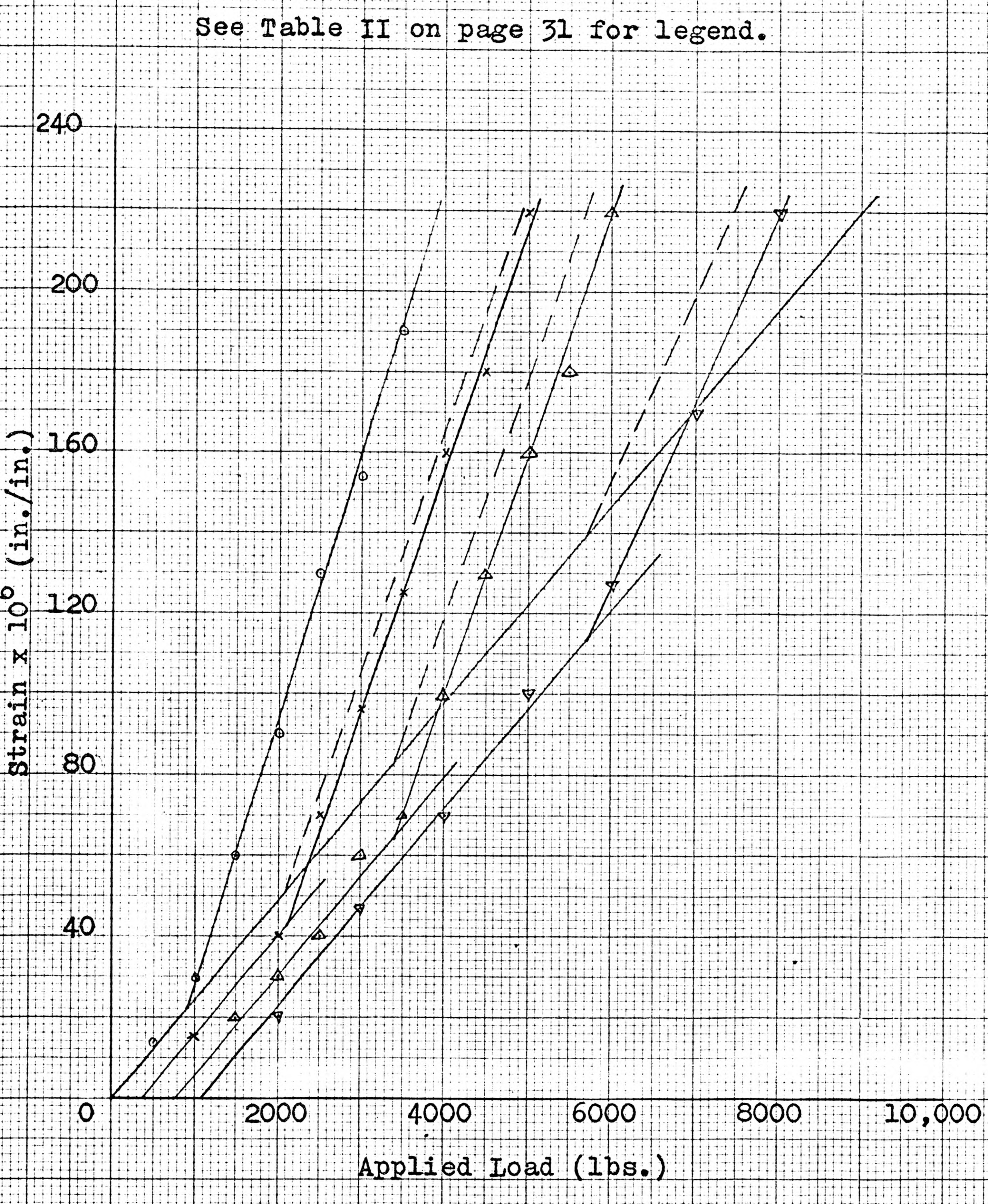

Figure 8. Tension Test with 3/8" Dia. Bolt

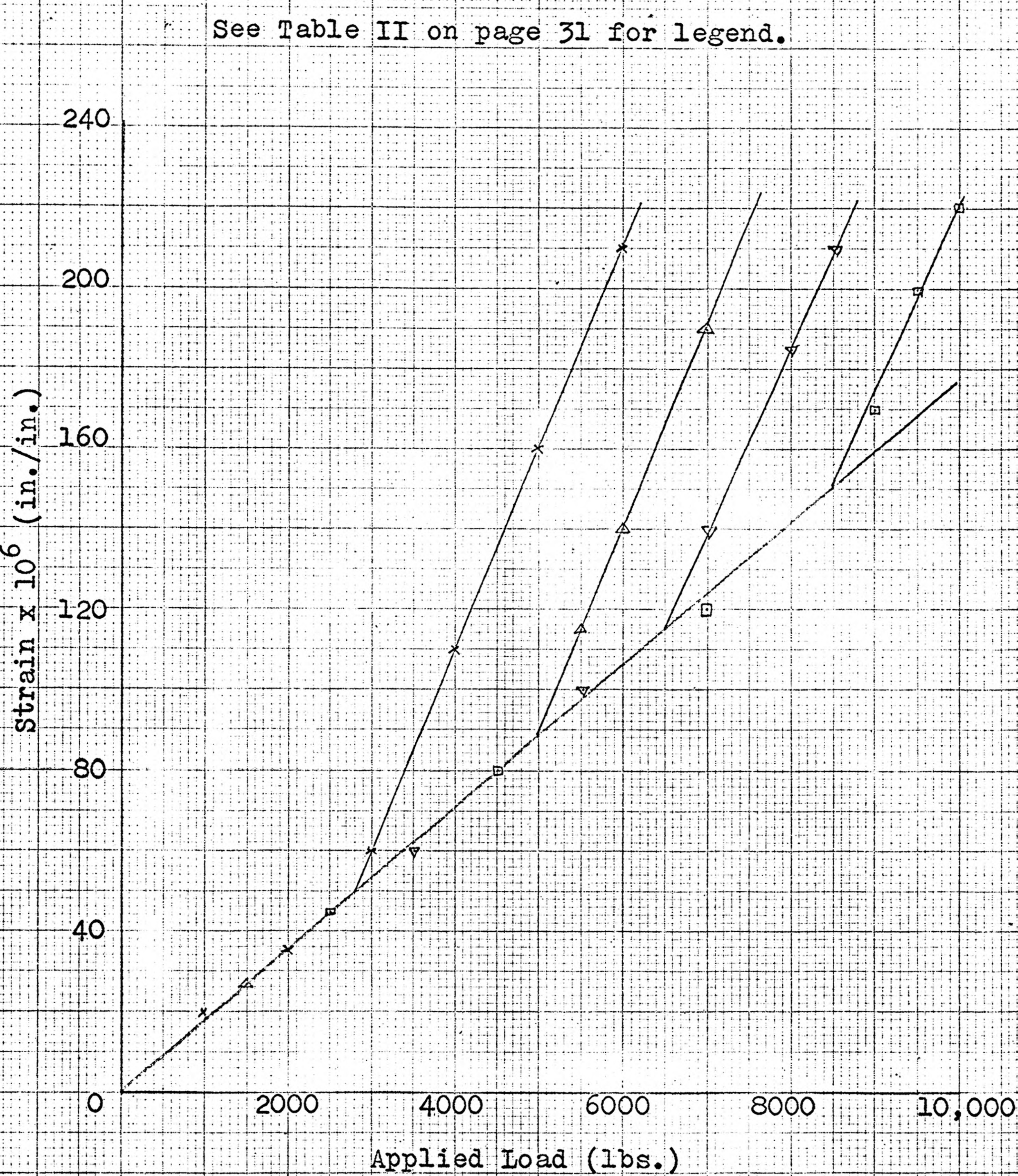

Figure 9. Tension Test with 1/2" Dia. Bolt

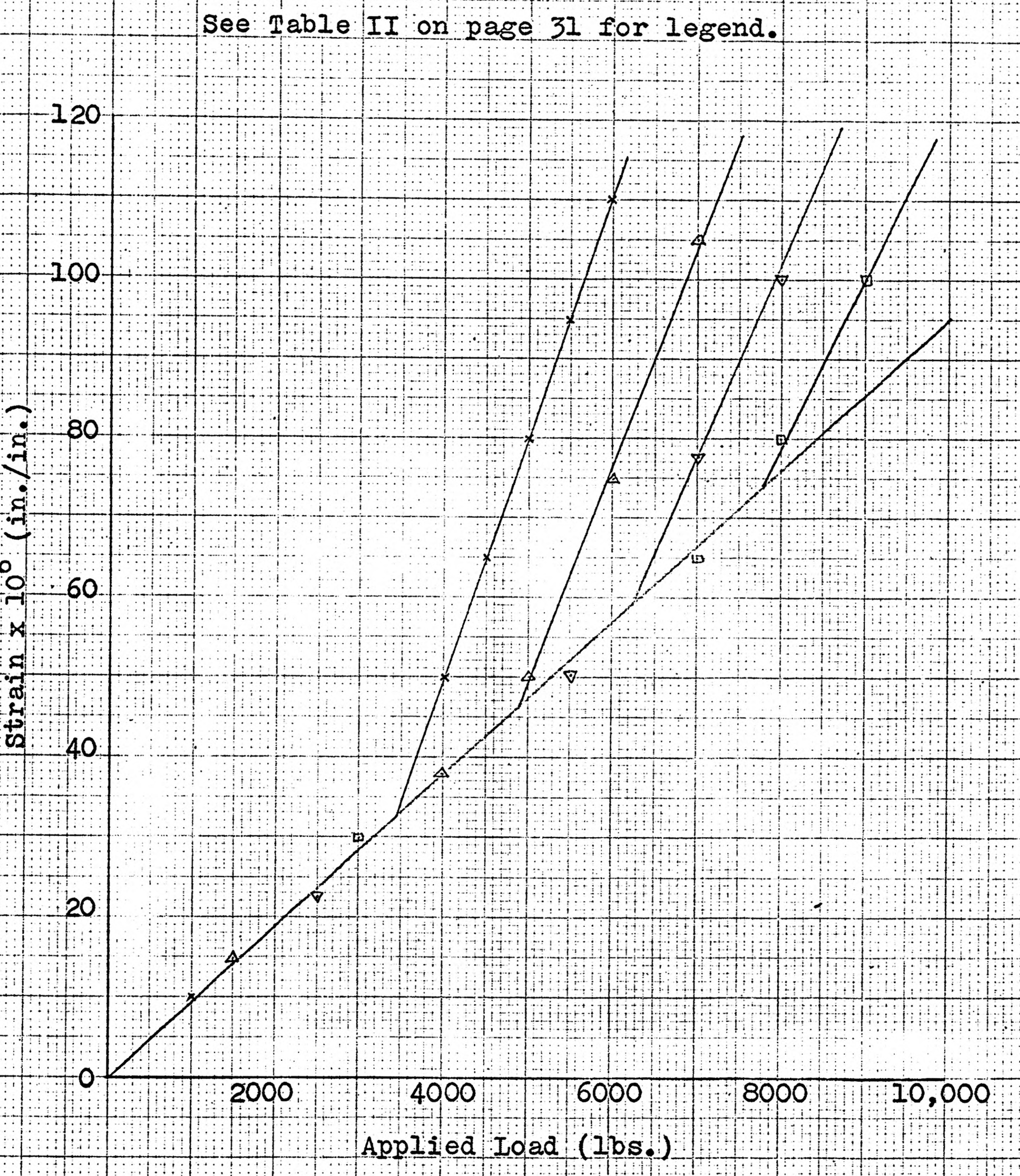

Figure 10. Tension Test with 5/8" Dia. Bolt

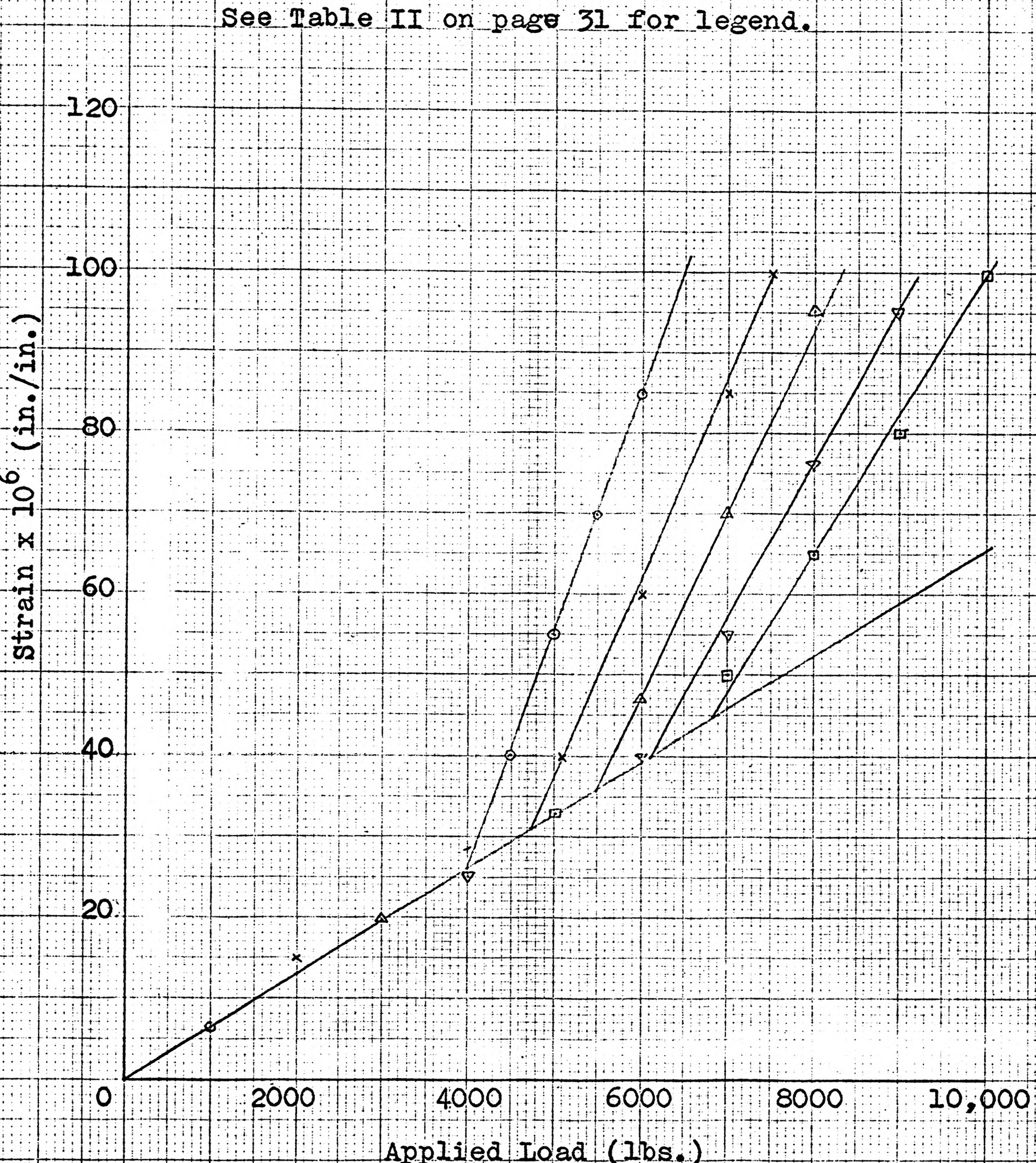

Figure 11. Tension Test with 3/4" Dia. Bolt

TABLE III

PREDICTED AND ACTUAL VALUES OF LOAD CAUSING SEPARATION

FOR DIFFERENT BOLT SIZES

Bolt Diameter:- 3/8"

Initial Torque (lb.in.)	Initial Strain $\times 10^6$ (in./in.)	Pretension (lbs.)	Load Causing Separation (lbs.) Calculated	Experimental
100	132	437	804	900
200	334	1,109	2,035	2,100
300	559	1,852	3,400	3,400
400	915	3,039	5,475	5,700
Bolt Diameter:- 1/2"				
200	262	1,545	2,990	2,800
300	438	2,580	5,000	5,000
400	613	3,615	7,000	6,500
500	793	4,675	9,040	8,500
Bolt Diameter:- 5/8"				
200	131	1,208	2,670	3,450
350	241	2,220	4,900	4,900
500	338	3,115	6,890	6,250
650	417	3,840	8,500	7,800
Bolt Diameter:- 3/4"				
350	84	1,113	2,800	4,000
450	133	1,765	4,430	4,750
550	163	2,161	5,440	5,500
650	183	2,425	6,100	6,100
750	214	2,840	7,125	6,850

2. BENDING TEST

In this experiment, the flange coupling was subjected to bending loads by the Universal Testing Machine. The arrangement of loading and the equipment are shown in Figure 12. The instrumented bolt was kept vertically below and furthest away from the point of loading. This is necessary to analyze the worst condition in bending. The supports were kept 24" apart. Hence, the bending moment at the center of the flange coupling was $6 \cdot P$, where P is the applied load at the center.

Table IV shows the sample data for a 3/8" diameter bolt. Similar data were obtained for 1/2", 5/8", and 3/4". From this data, the graphs of applied bending moment versus strain were plotted. The bending moment causing separation of the flanges was obtained when a sudden change in slope was observed, as shown in Figure 13. All graphs were corrected to the same origin. Sample calculations for the 3/8" diameter bolts are given on page 42.

Figure 12. The Flange Coupling in Bending Test.

TABLE IV

SAMPLE DATA FOR BENDING TEST

Flange Diameter:-	6.5"
Flange Thickness:-	5/8"
Bolt Circle Diameter:-	4.5"
Bolt Diameter:-	3/8"
Number of Bolts:-	3
Initial Meter Reading:-	11,430

INITIAL TIGHTENING TORQUE:- 100 in. lb.

Load (lbs.)	Bending Moment (lb.in.)	Meter Reading	Strain x 10^6 (in./in.)
0	0	11.627	0
100	600	11,632	5
200	1,200	11,642	15
300	1,800	11,657	30
400	2,400	11,672	45
500	3,000	11,687	60
600	3,600	11,697	70
700	4,200	11,712	85

TABLE IV Continued

INITIAL TIGHTENING TORQUE:- 200 in. lb.

Load (lbs.)	Bending Moment (lb.in.)	Meter Reading	Strain x 10^6(in./in.)
0	0	11,950	0
200	1,200	11,957	7
400	2,400	11,965	15
500	3,000	11,975	25
600	3,600	11,990	40
700	4,200	12,005	55
800	4,800	12,020	70

INITIAL TIGHTENING TORQUE:- 300 in. lb.

Load (lbs.)	Bending Moment (lb.in.)	Meter Reading	Strain x 10^6(in./in.)
0	0	12,258	0
200	1,200	12,263	5
300	1,800	12,268	10
500	3,000	12,278	20
700	4,200	12,288	30
900	5,400	12,308	50
1000	6,000	12,318	60
1100	6,600	12,328	70
1200	7,200	12,343	85

TABLE IV Continued

INITIAL TIGHTENING TORQUE:- 400 in. lb.

Load (lbs.)	Bending Moment (lb.in.)	Meter Reading	Strain x 10^6 (in./in.)
0	0	12,478	0
300	1,800	12,483	5
400	2,400	12,488	10
600	3,600	12,498	20
800	4,800	12,508	30
1,000	6,000	12,523	45
1,100	6,600	12,533	55
1,200	7,200	12,543	65
1,300	7,800	12,553	75

Sample Calculations For Bending Tests

Using 3/8" Diameter Bolt,

From Figure 13, at 300 in. lb. torque, the bending moment causing separation of the flanges is 4,320 in. lb. The initial strain in the bolts was 828×10^{-6} in./in.

From equation 25,

$$M = \frac{\epsilon_{bi} \cdot E_b \cdot A_b \cdot D_f^2}{8 \cdot D_b \cdot K_3}$$

$$4{,}320 = \frac{828 \times 10^{-6} \times 30 \times 10^6 \times \frac{\pi}{4} (3/8)^2 \times 6.5^2}{8 \times 4.5 \times K_3}$$

So, $\underline{K_3 = 0.745}$

Hence,

$$M = \frac{\epsilon_{bi} \times 30 \times 10^6 \times \frac{\pi}{4} (3/8)^2 \times 6.5^2}{8 \times 4.5 \times 0.745}$$

$$\underline{M = 5.225 \times 10^6 \times \epsilon_{bi}}$$

Table VI shows the predicted and actual values of the bending moment causing separation of the flanges for different bolt sizes at different initial tension in the bolts.

TABLE V

LEGEND FOR BENDING TEST GRAPHS

	Initial Tightening Torque
—⊙—⊙—⊙—	100 in. lb.
—×—×—×—	200 in. lb.
—△—△—△—	300 in. lb.
—▽—▽—▽—	400 in. lb.
—⊡—⊡—⊡—	500 in. lb.

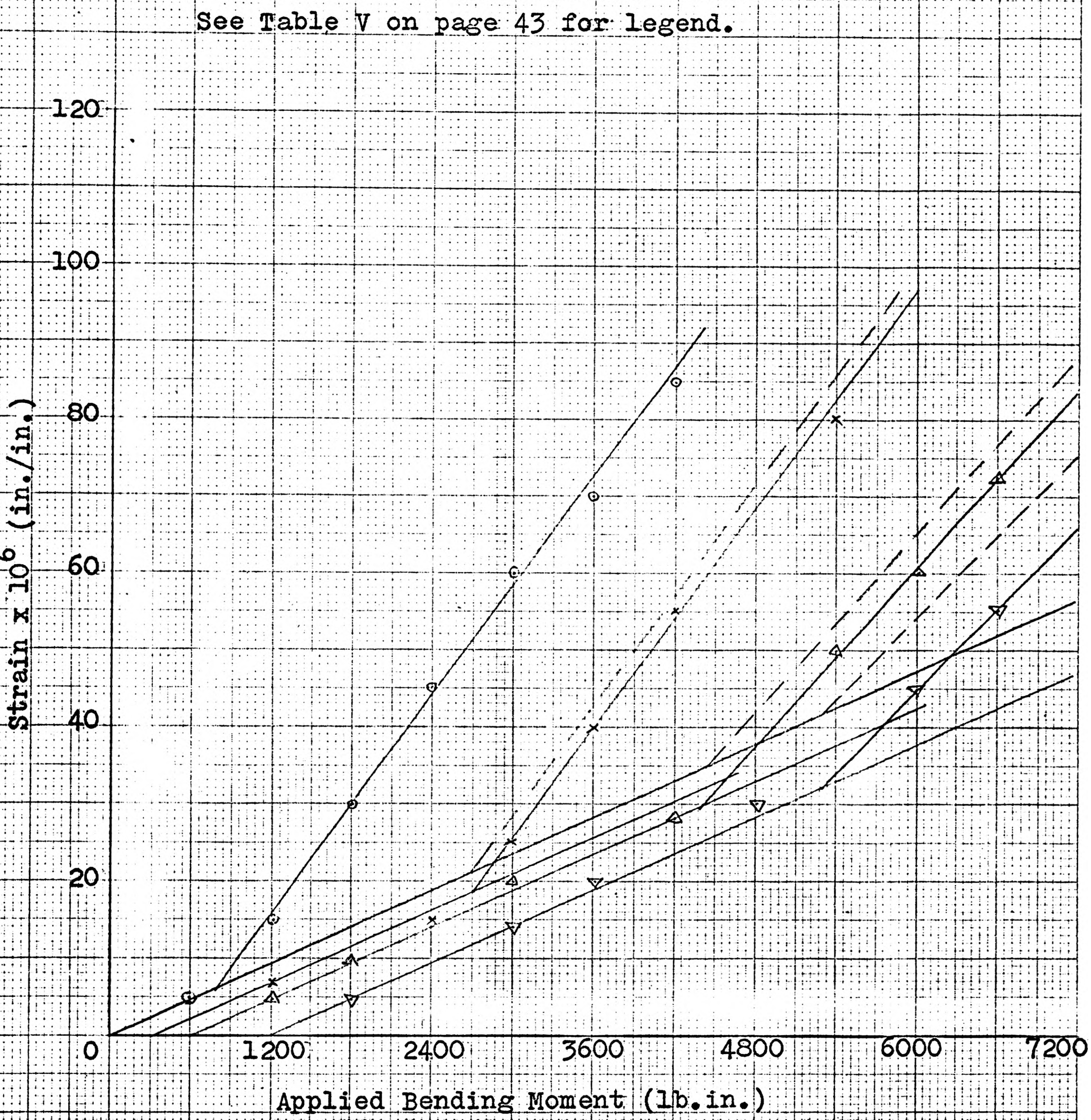

Figure 13. Bending Test with 3/8" Dia. Bolt

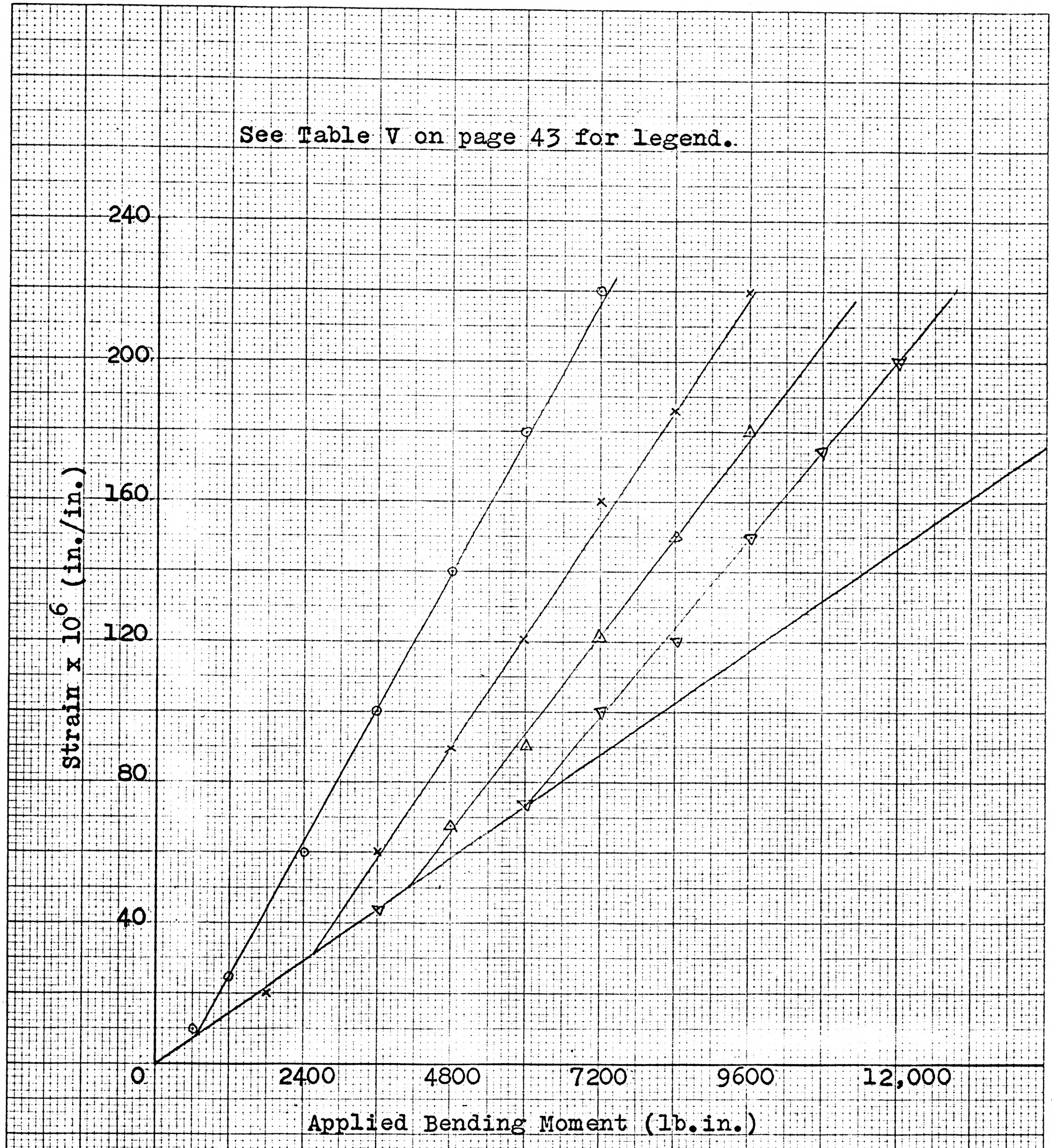

Figure 14. Bending Test with 1/2" Dia. Bolt

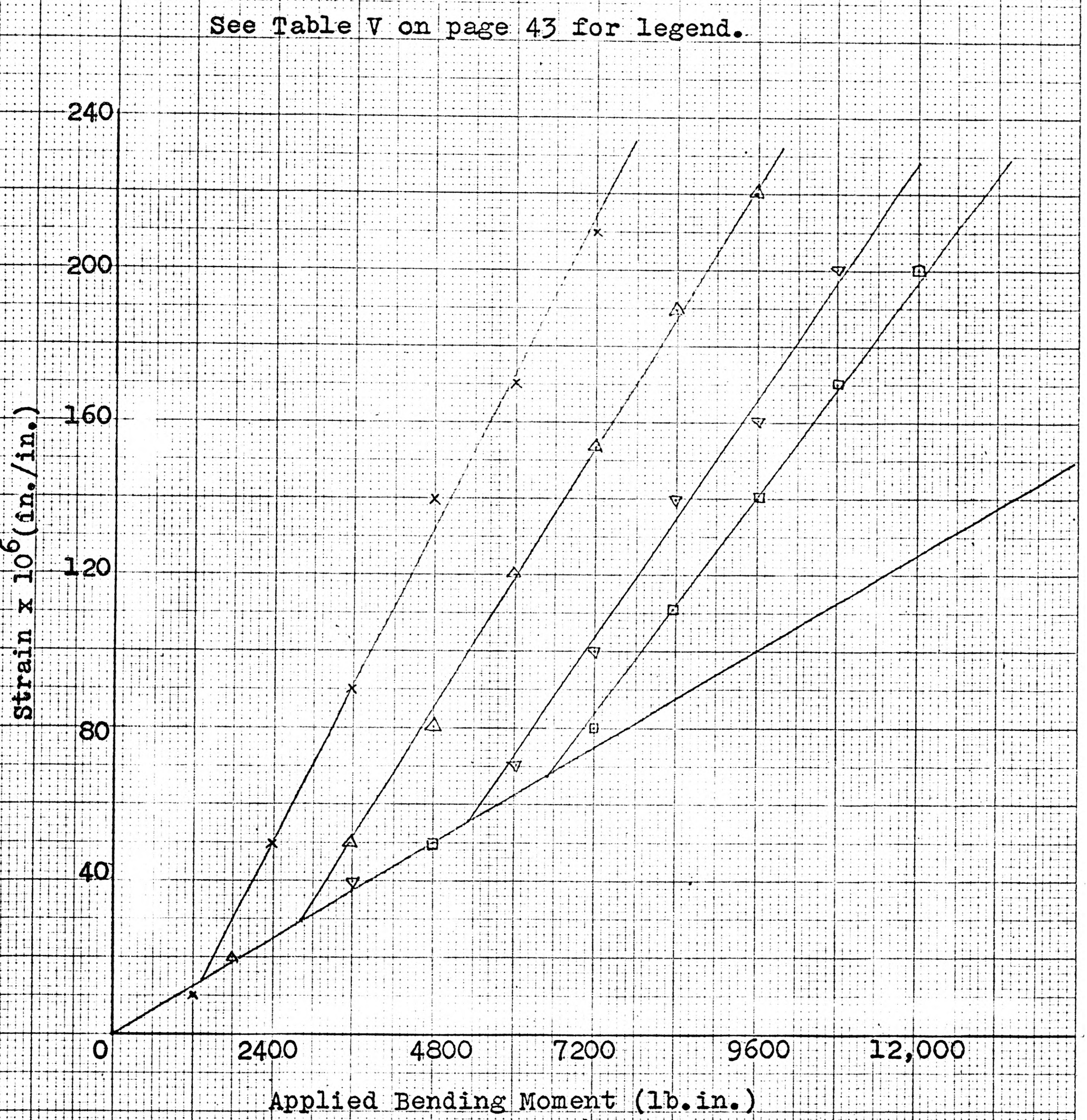

Figure 15. Bending Test with 5/8" Dia. Bolt

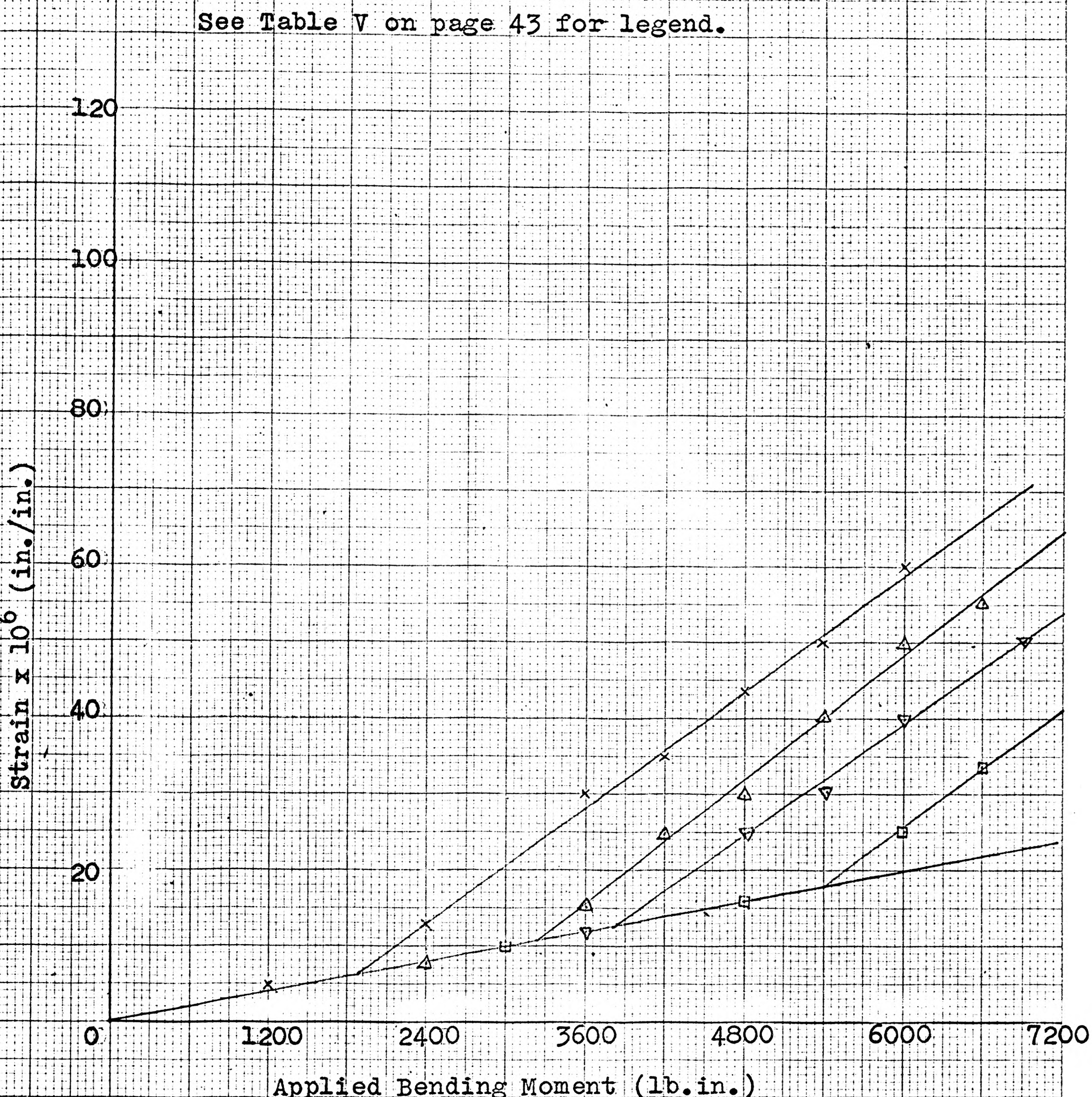

Figure 16. Bending Test with 3/4" Dia. Bolt

TABLE VI

PREDICTED AND ACTUAL VALUES OF BENDING MOMENT CAUSING SEPARATION

FOR DIFFERENT BOLT SIZES

Initial Torque (in. lb.)	Initial Strain x 10^6 (in./in.)	Pretension (lbs.)	Bending Moment Causing Separation (lb. in.) Calculated	Experimental
Bolt Diameter:- 3/8"				
100	197	653	1,030	780
200	520	1,725	2,720	2,700
300	828	2,741	4,320	4,320
400	1,048	3,470	5,470	5,280
Bolt Diameter:-1/2"				
100	92	543	851	600
200	250	1,475	2,319	2,520
300	430	2,538	3,980	4,080
400	649	3,825	6,000	6,000
Bolt Diameter:- 5/8"				
200	94	865	1,345	1,290
350	220	2,022	3,140	2,820
500	370	3,410	5,280	5,280
650	438	4,040	6,260	6,480
Bolt Diameter:- 3/4"				
200	78	1,035	1,620	1,860
350	158	2,100	3,280	3,150
500	196	2,600	4,075	3,780
650	245	3,250	5,100	5,100

B. Testing of Flange Coupling with Varying Flange Diameter.

In this experiment, a pair of flanges of 8.0" diameter were prepared. Three bolts of 1/2" diameter were used as fasteners. The flange coupling was tested in tension and bending according to the procedure of the previous experiments. The flanges of 8.0", 7.5", 7.0", 6.5", and 6.0" diameter were tested. The following pages show the graphs obtained in the tests.

Tables VII and VIII show the predicted and actual values of load and bending moment causing separation of the flanges.

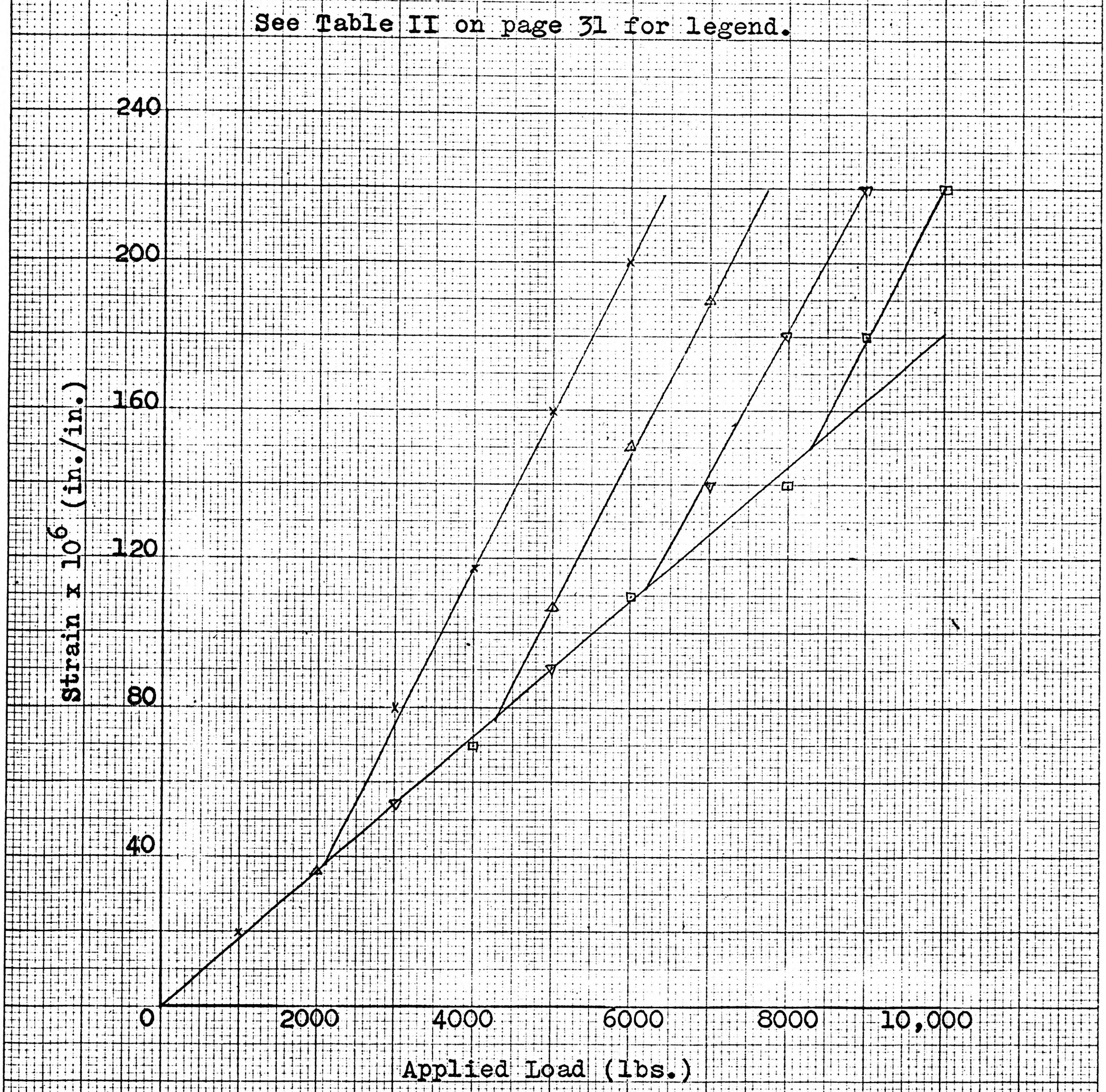

Figure 17. Tension Test with 8.0" Dia. Flange

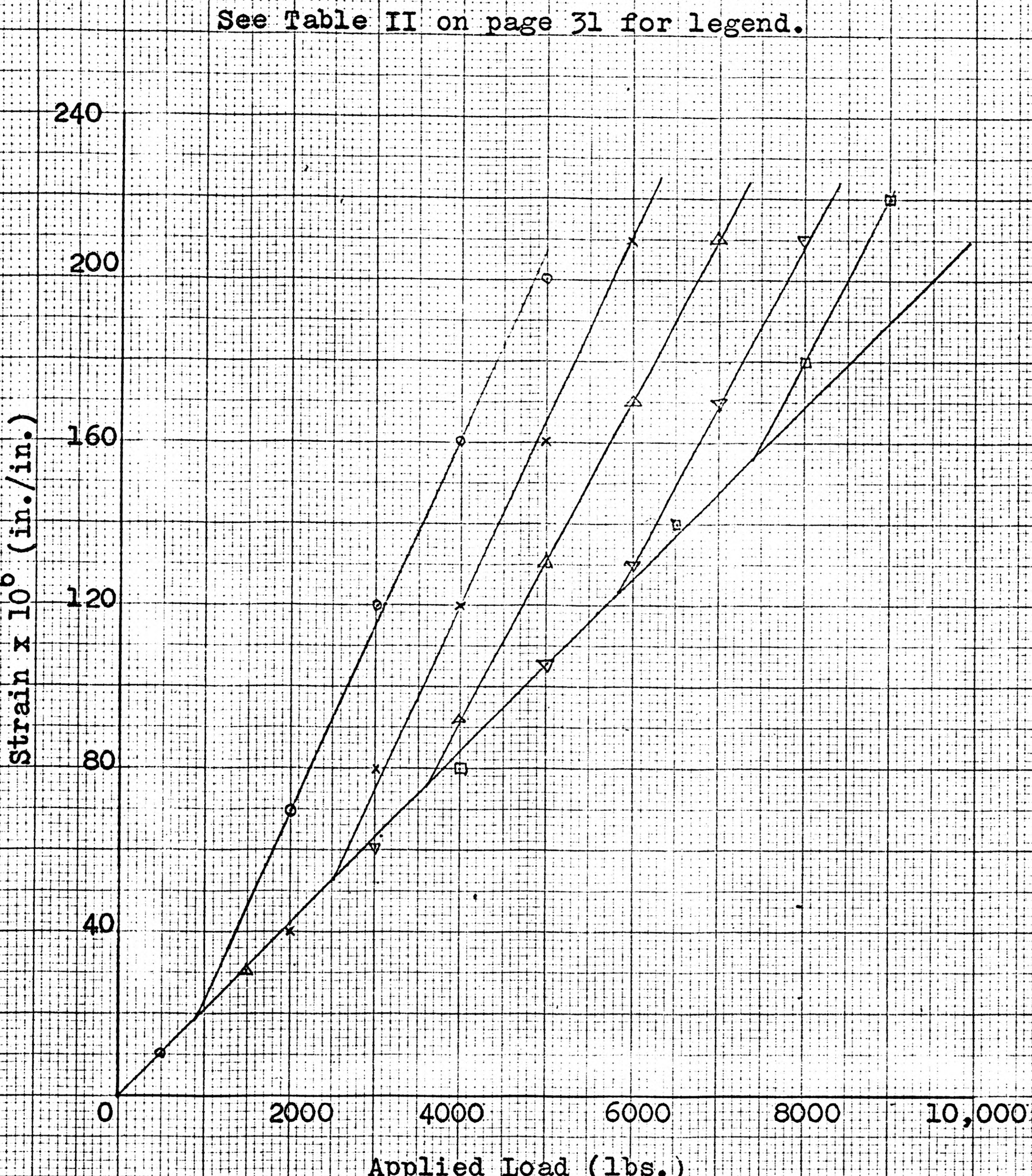

Figure 18. Tension Test with 7.5" Dia. Flange

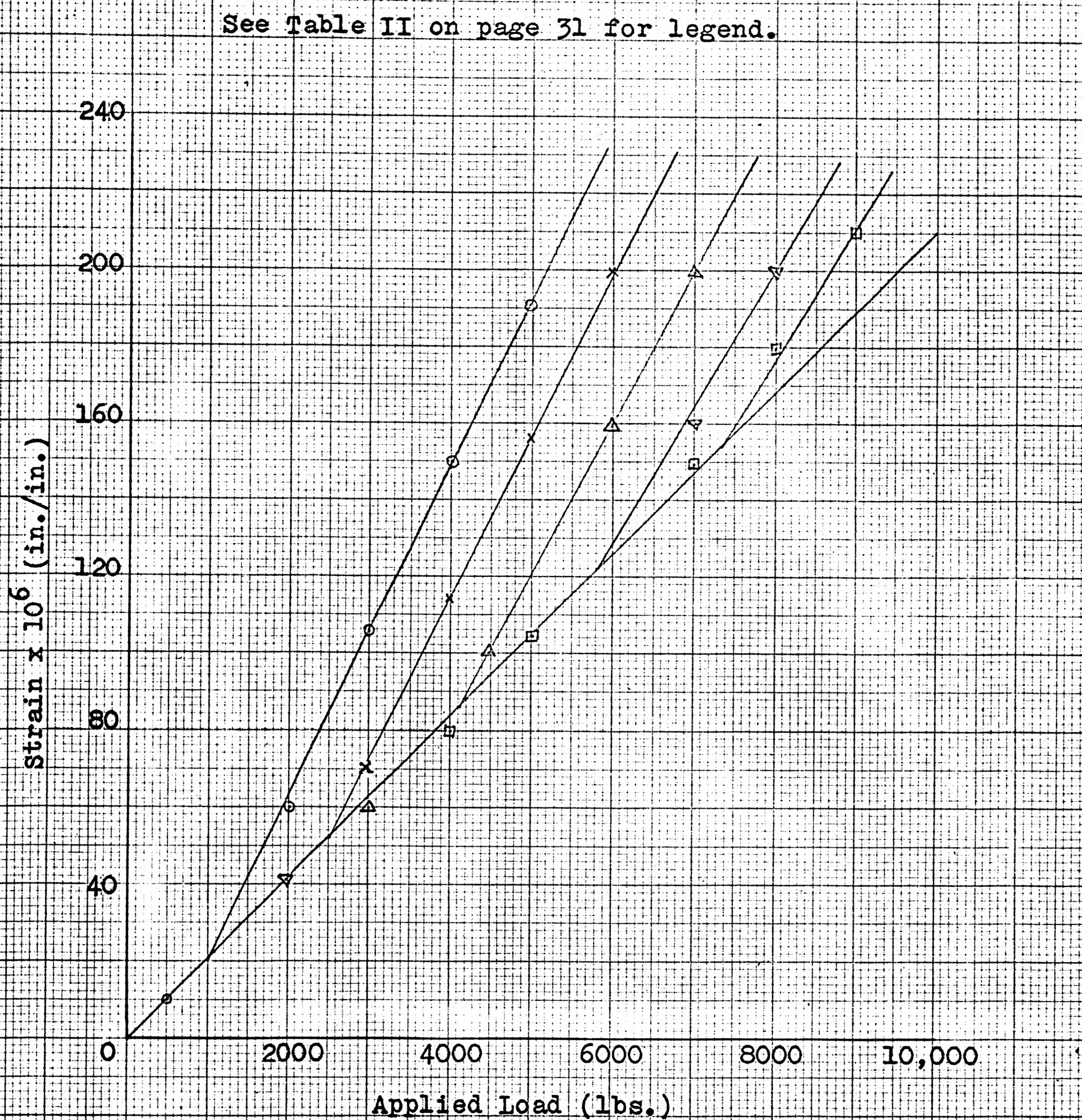

Figure 19. Tension Test with 7.0" Dia. Flange

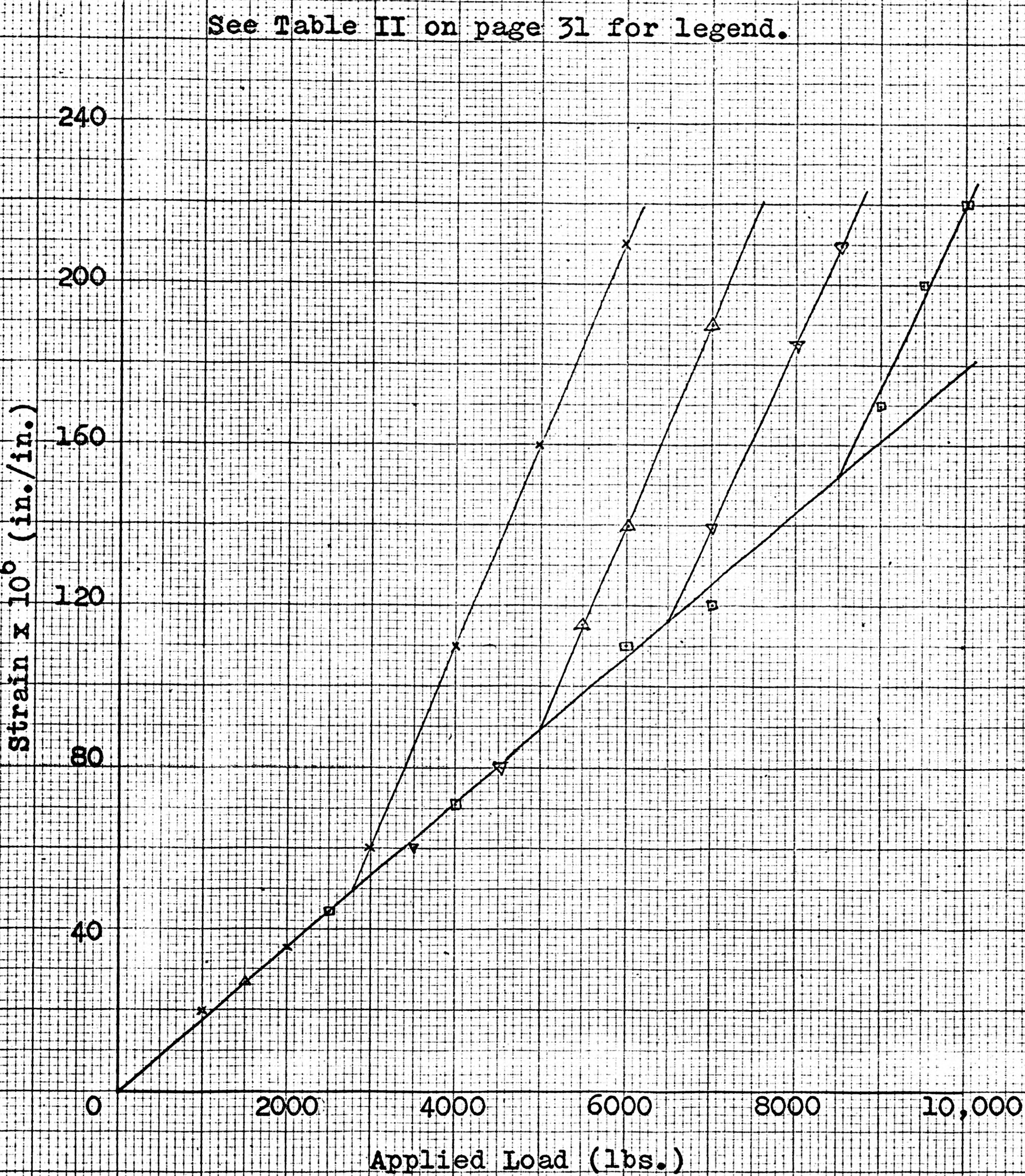

Figure 20. Tension Test with 6.5" Dia. Flange

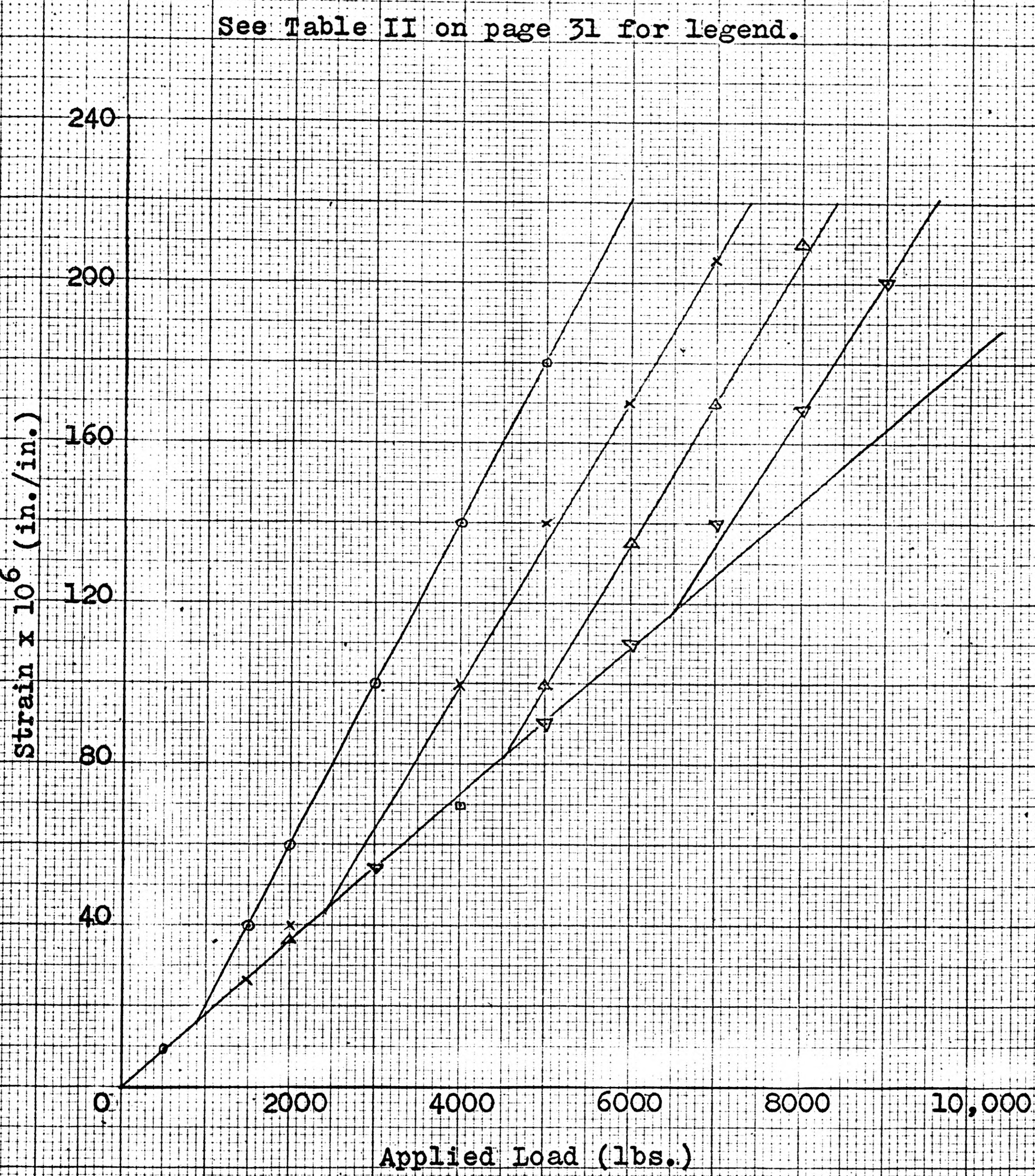

Figure 21. Tension Test with 6.0" Dia. Flange

TABLE VII

PREDICTED AND ACTUAL VALUES OF LOAD CAUSING SEPARATION OF FLANGES

FOR DIFFERENT FLANGE SIZES

Initial Torque (in. lb.)	Initial Strain $\times 10^6$ (in./in.)	Pretension (lbs.)	Load Causing Separation (lbs.) Calculated	Load Causing Separation (lbs.) Experimental
Flange Diameter:- 8.0"				
200	211	1,242	2,370	2,100
300	382	2,250	4,300	4,300
400	580	3,420	6,515	6,200
500	715	4,210	8,040	8,300
Flange Diameter:- 7.5"				
100	91	536	819	900
200	263	1,550	2,347	2,500
300	400	2,361	3,600	3,600
400	606	3,570	5,454	5,850
500	741	4,360	6,669	7,400
Flange Diameter:- 7.0"				
100	96	566	927	1,000
200	279	1,642	2,695	2,500
300	426	2,510	4,100	4,100
400	594	3,500	5,830	5,800
500	800	4,710	7,720	7,350

TABLE VII Continued

Flange Diameter:- 6.5"

Initial Torque (in. lb.)	Initial Strain $\times 10^6$ (in./in.)	Pretension (lbs.)	Load Causing Separation (lbs.) Calculated	Experimental
200	262	1,545	2,990	2,800
300	438	2,580	5,000	5,000
400	613	3,615	7,000	6,500
500	793	4,675	9,040	8,500
	Flange Diameter:- 6.0"			
100	67	395	792	900
200	201	1,184	2,400	2,400
300	397	2,340	4,700	4,500
400	585	3,445	6,925	6,450

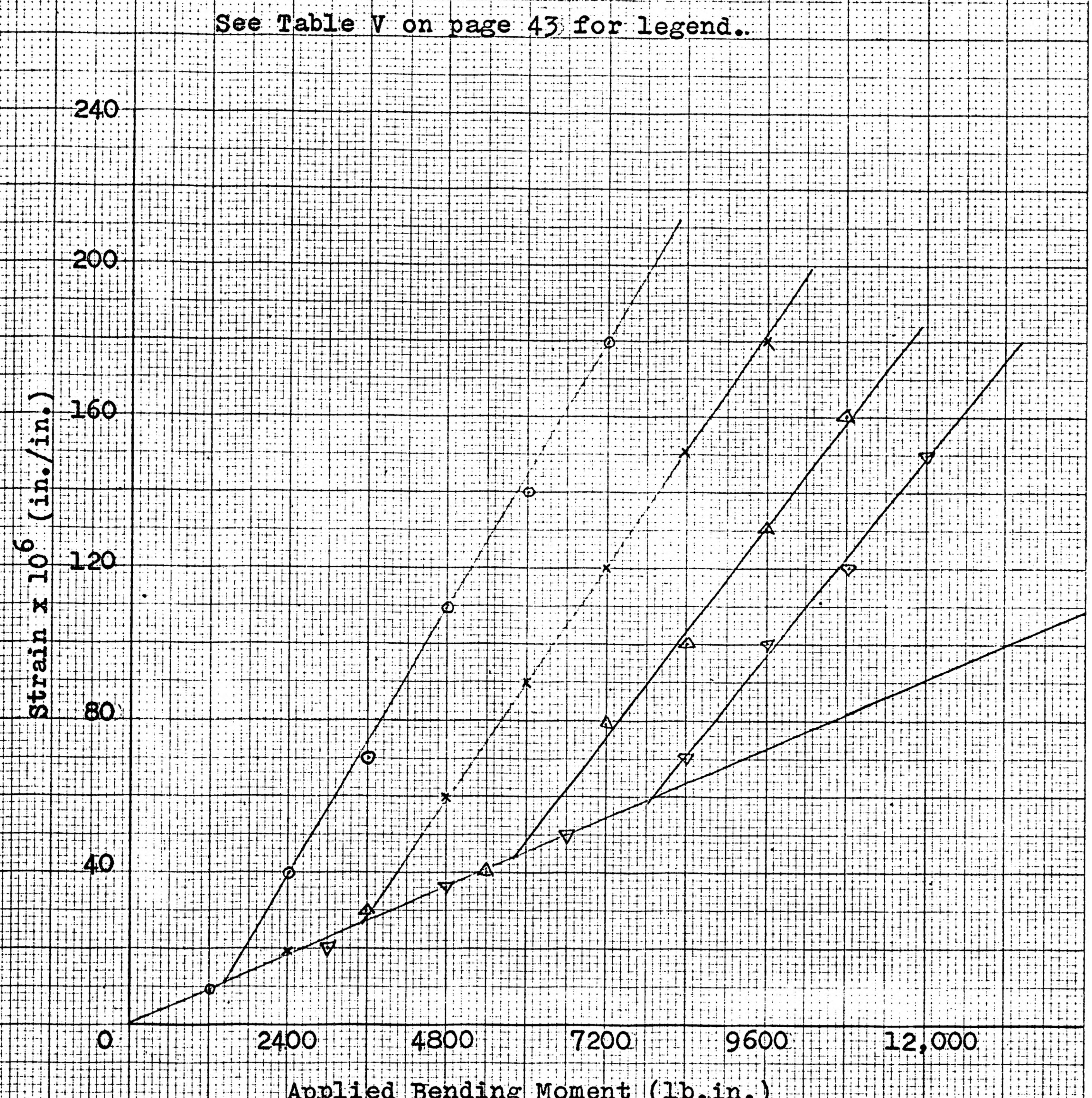

Figure 22. Bending Test with 8.0" Dia. Flange

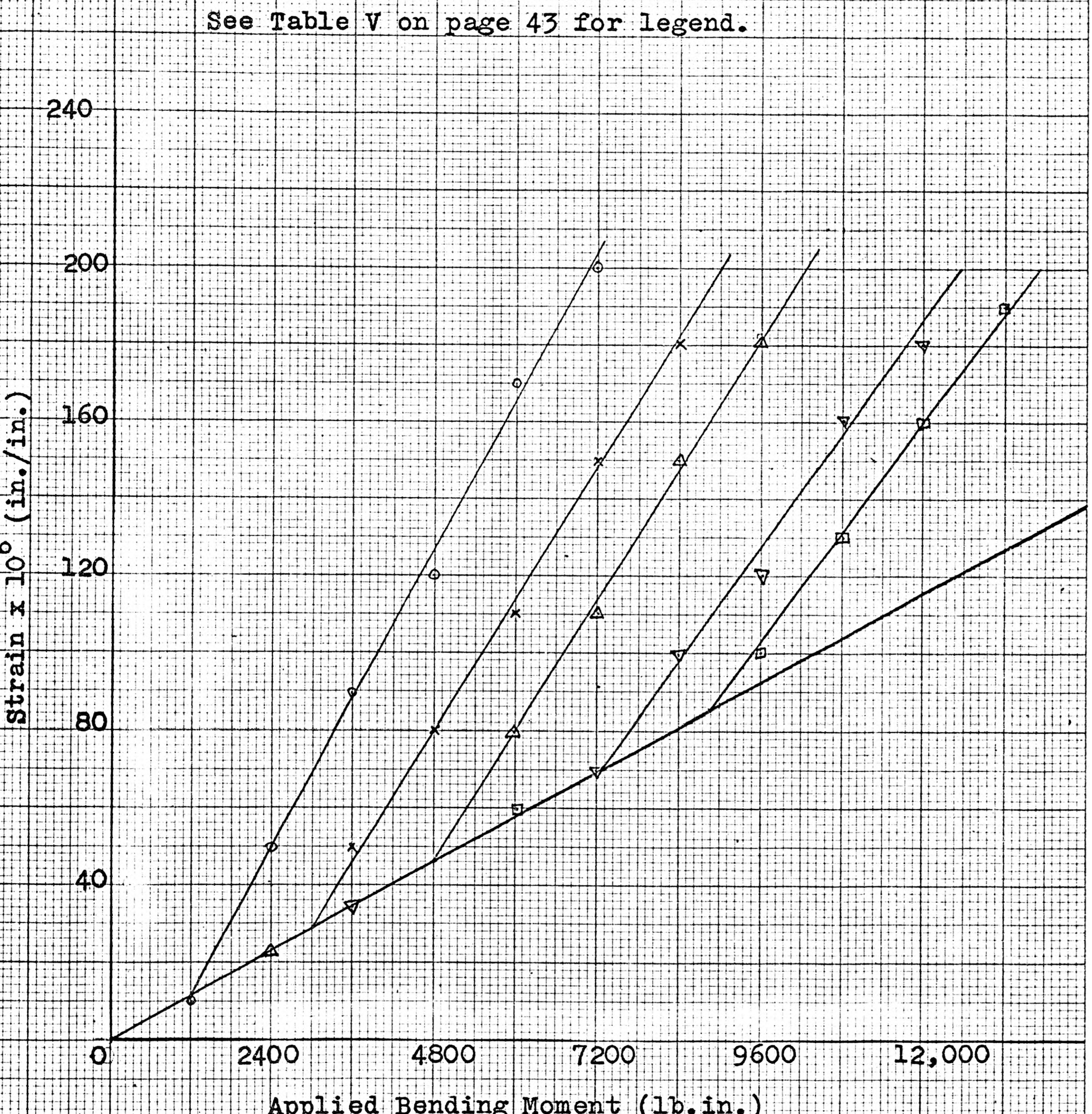

Figure 23. Bending Test with 7.5" Dia. Flange

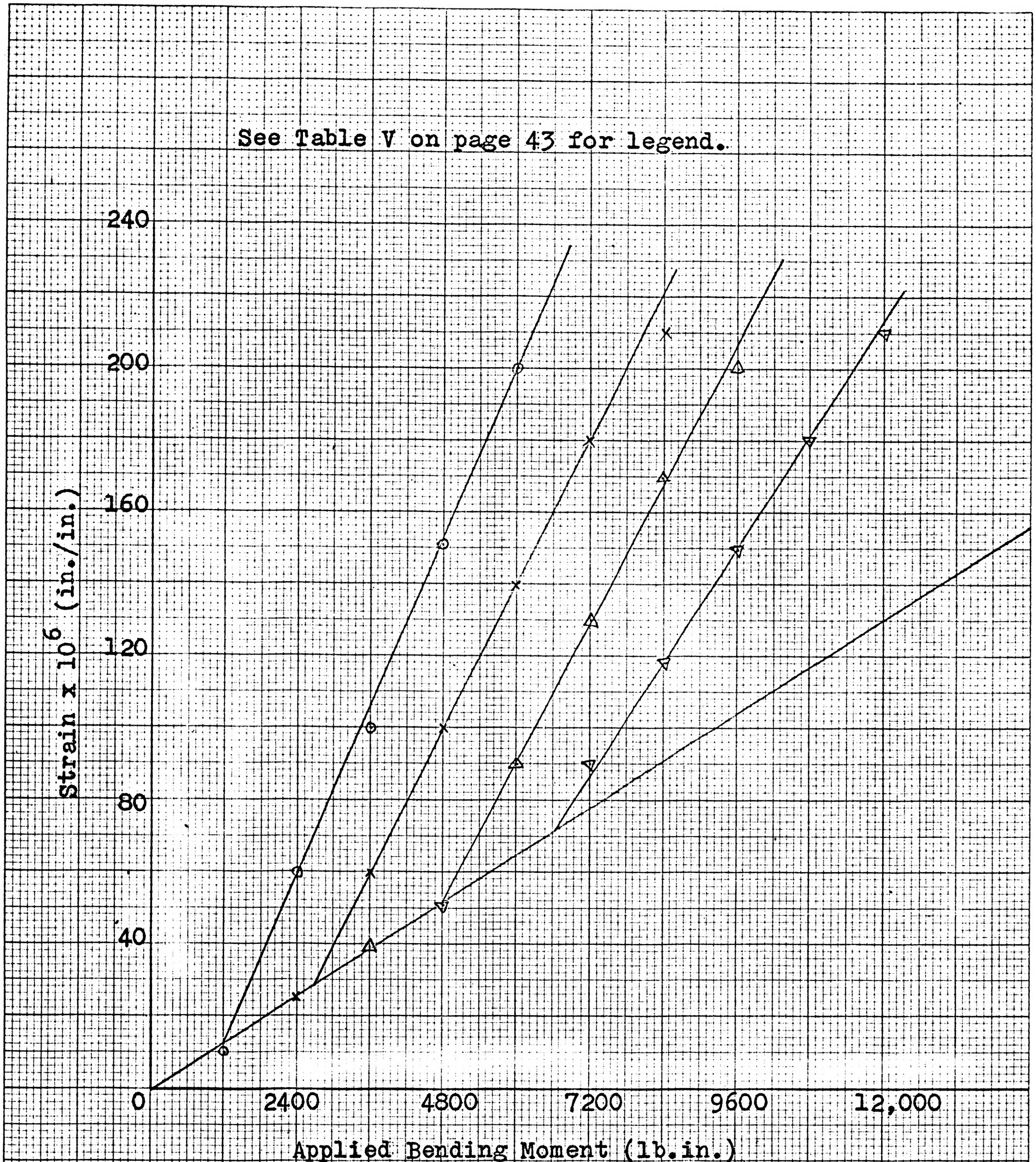

Figure 24. Bending Test with 7.0" Dia. Flange

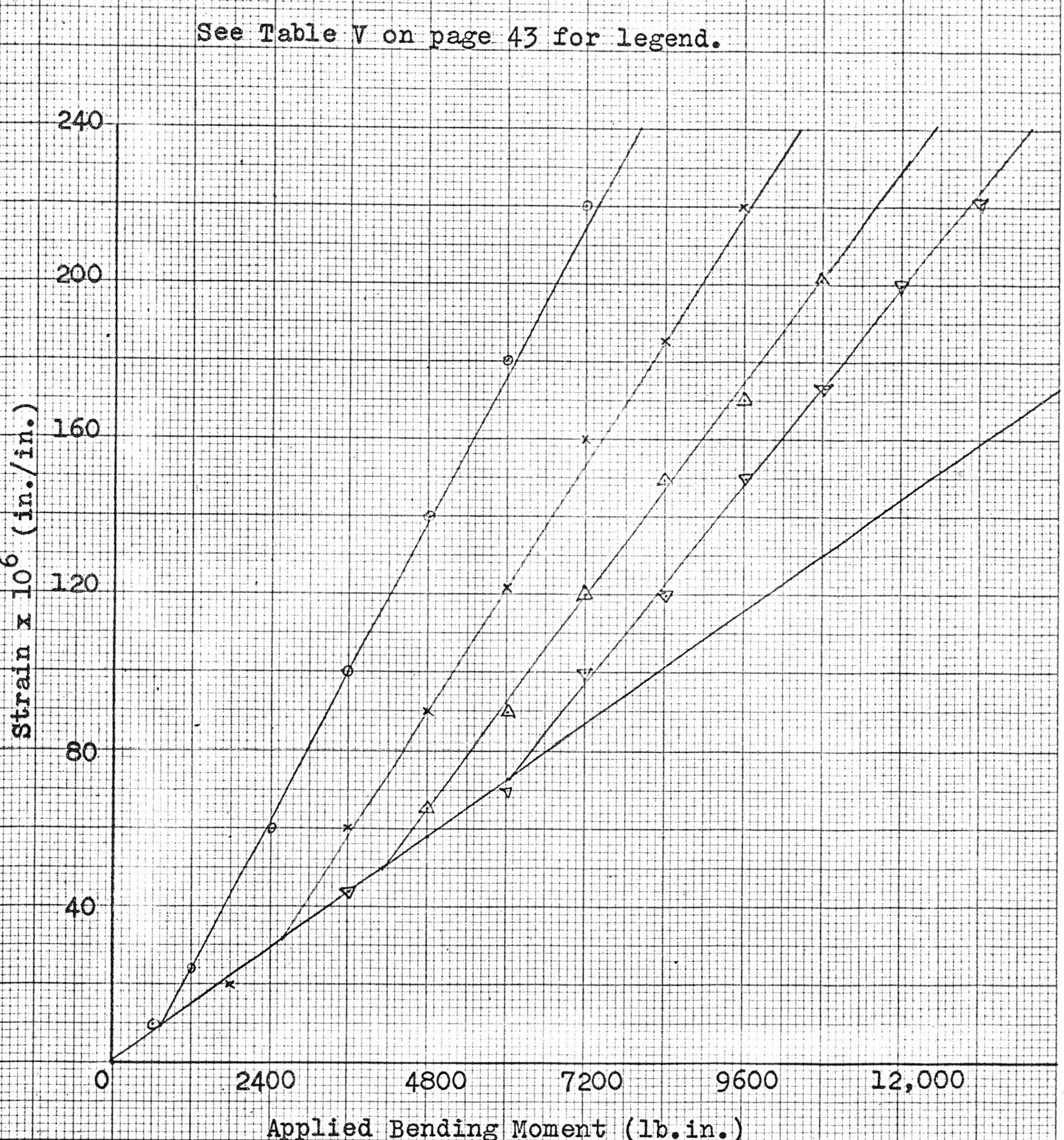

Figure 25. Bending Test with 6.5" Dia. Flange

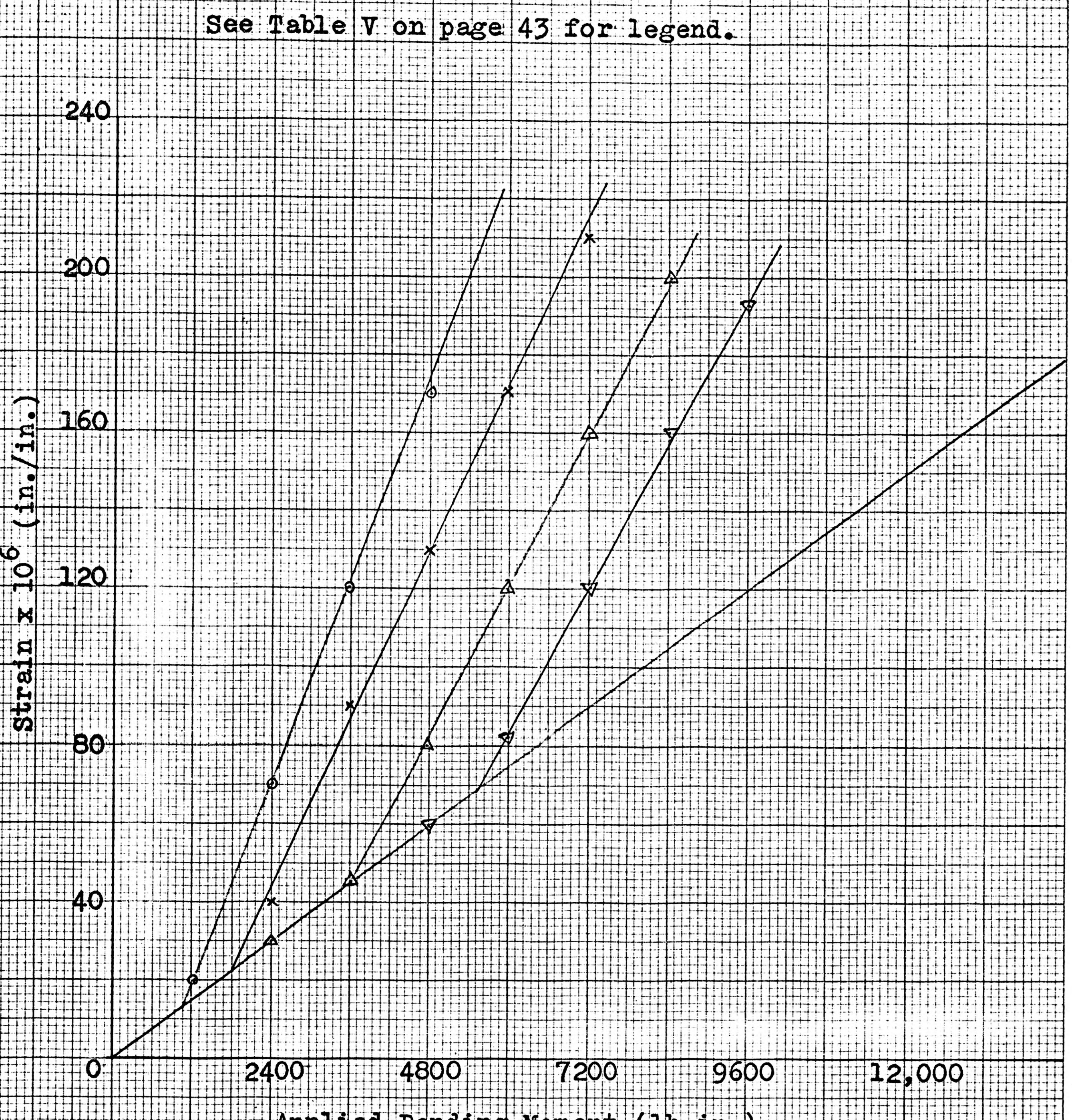

Figure 26. Bending Test with 6.0" Dia. Flange

TABLE VIII

PREDICTED AND ACTUAL VALUES OF BENDING MOMENT CAUSING SEPARATION

FOR DIFFERENT FLANGE SIZES

Initial Torque (in. lb.)	Initial Strain $\times 10^6$ (in./in.)	Pretension (lbs.)	Bending Moment Causing Separation (lb. in.) Calculated	Bending Moment Causing Separation (lb. in.) Experimental
Flange Diameter:- 8.0"				
100	92	543	1,240	1,440
200	240	1,412	3,239	3,600
300	398	2,342	5,370	5,820
400	578	3,400	7,800	7,800
Flange Diameter:- 7.5"				
100	98	577	1,175	1,200
200	248	1,460	2,970	3,000
300	420	2,475	5,035	4,800
400	601	3,540	7,200	7,200
500	700	4,120	8,390	8,820
Flange Diameter:- 7.0"				
100	124	730	1,316	1,200
200	290	1,709	3,080	2,700
300	450	2,650	4,770	4,800
400	621	3,660	6,600	6,600

TABLE VIII Continued

Flange Diameter:- 6.5"

Initial Torque (in. lb.)	Initial Strain $\times 10^6$ (in./in.)	Pretension (lbs.)	Bending Moment Causing Separation (lb. in.)	
			Calculated	Experimental
100	92	543	851	600
200	250	1,475	2,319	2,520
300	430	2,538	3,980	4,080
400	649	3,825	6,000	6,000

Flange Diameter:- 6.0"

Initial Torque (in. lb.)	Initial Strain $\times 10^6$ (in./in.)	Pretension (lbs.)	Calculated	Experimental
100	62	365	569	1,020
200	221	1,300	2,025	1,800
300	390	2,295	3,482	3,600
400	589	3,462	5,400	5,400

C. Testing of Flange Coupling With Gaskets:-

Gaskets are widely used to prevent migration of fluids, gases, or particles across the joint (4). The purpose of this experiment was to determine the influence of gaskets on the strength of a bolted joint. There are innumerable types of gasket materials in use today. Rubber and Copper gaskets of 1/8" thickness were tested in tension and bending according to the procedure of the previous experiments. Copper gaskets of 1/4", and 3/8" thickness were also tested. The following pages show the graphs obtained in the tests.

Tables IX and X show the predicted and actual values of load and bending moment causing separation of the flanges.

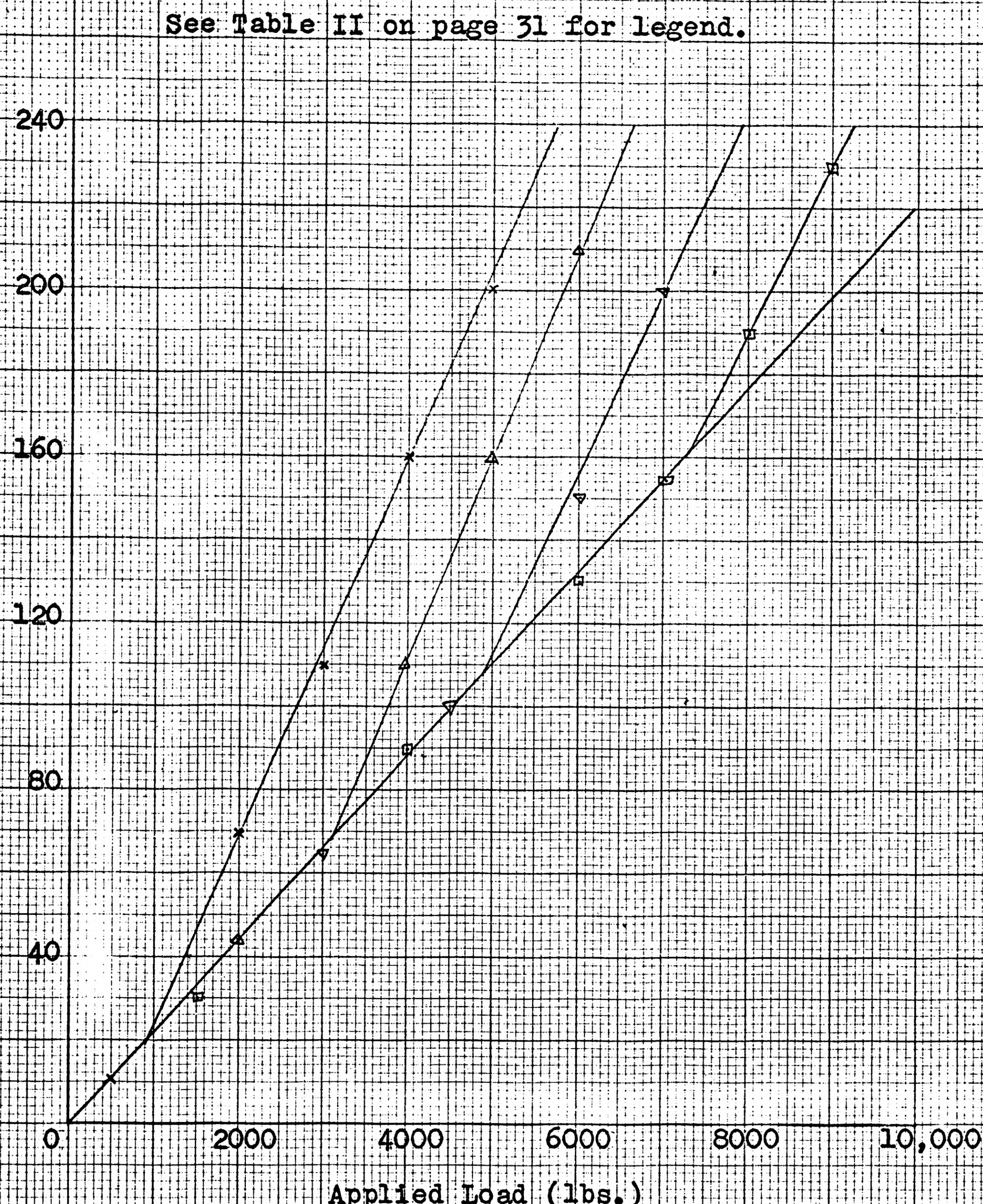

Figure 27. Tension Test with 1/8" Thick Rubber Gasket

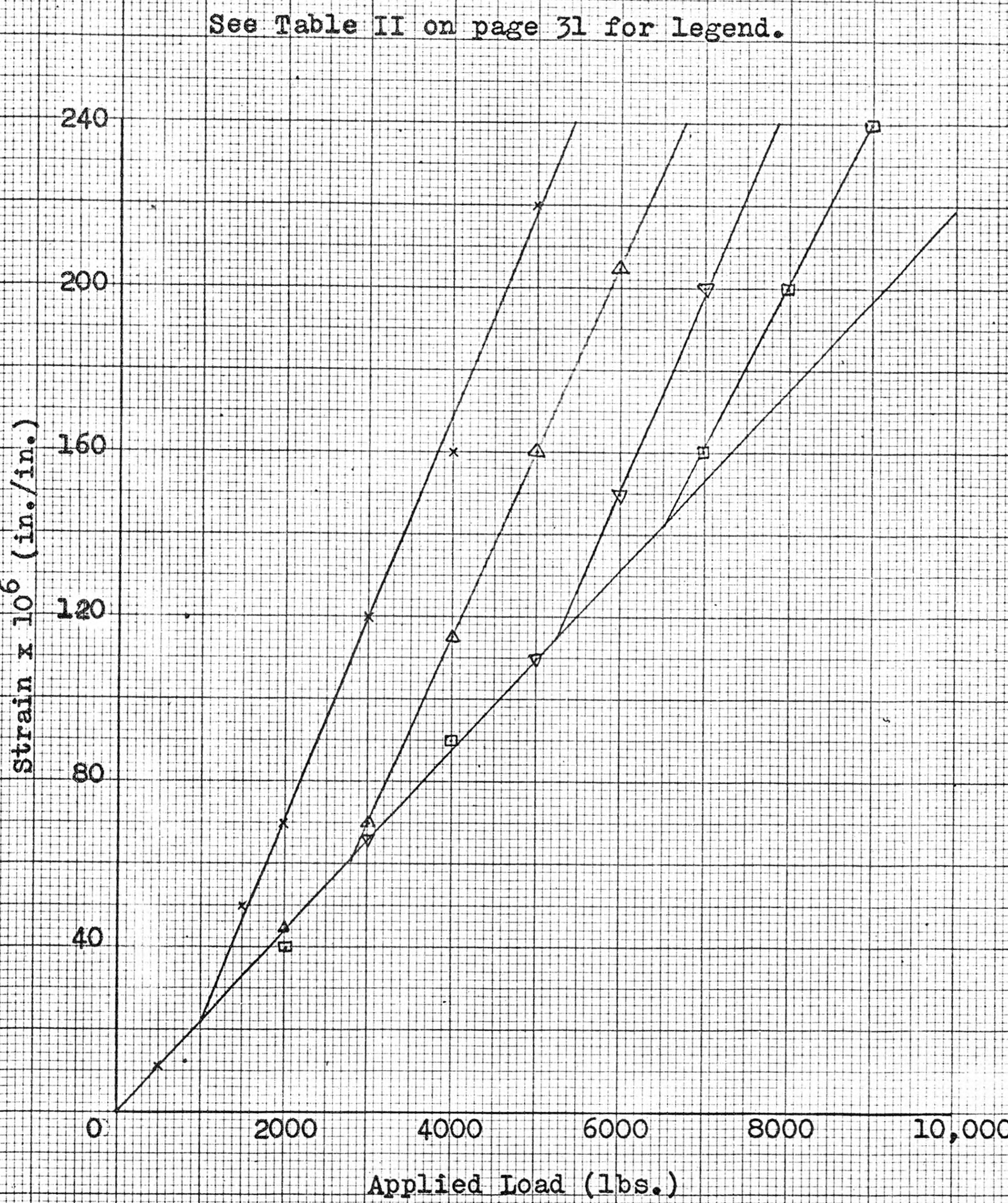

Figure 28. Tension Test with 1/8" Thick Copper Gasket

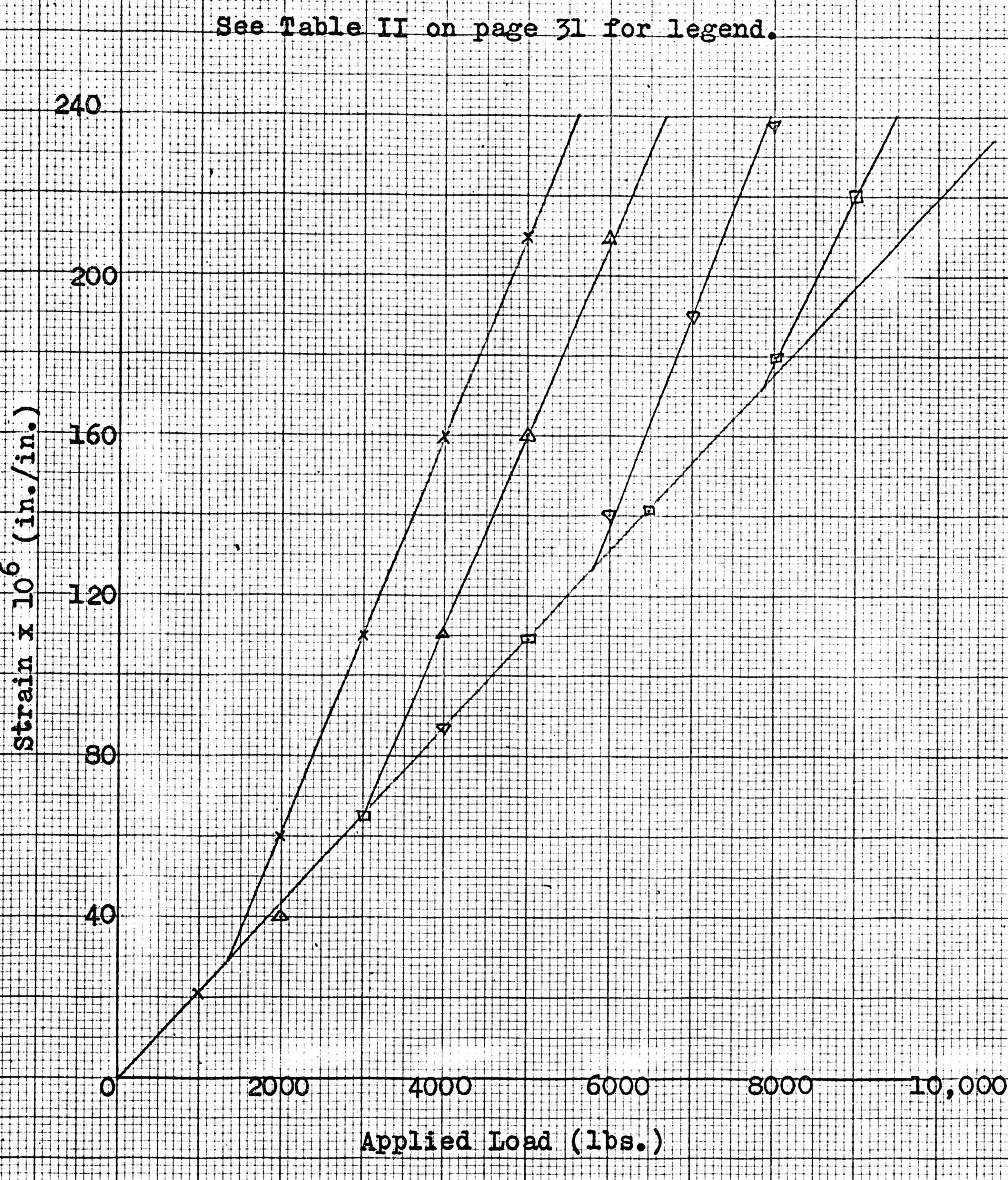

Figure 29. Tension Test with 1/4" Thick Copper Gasket

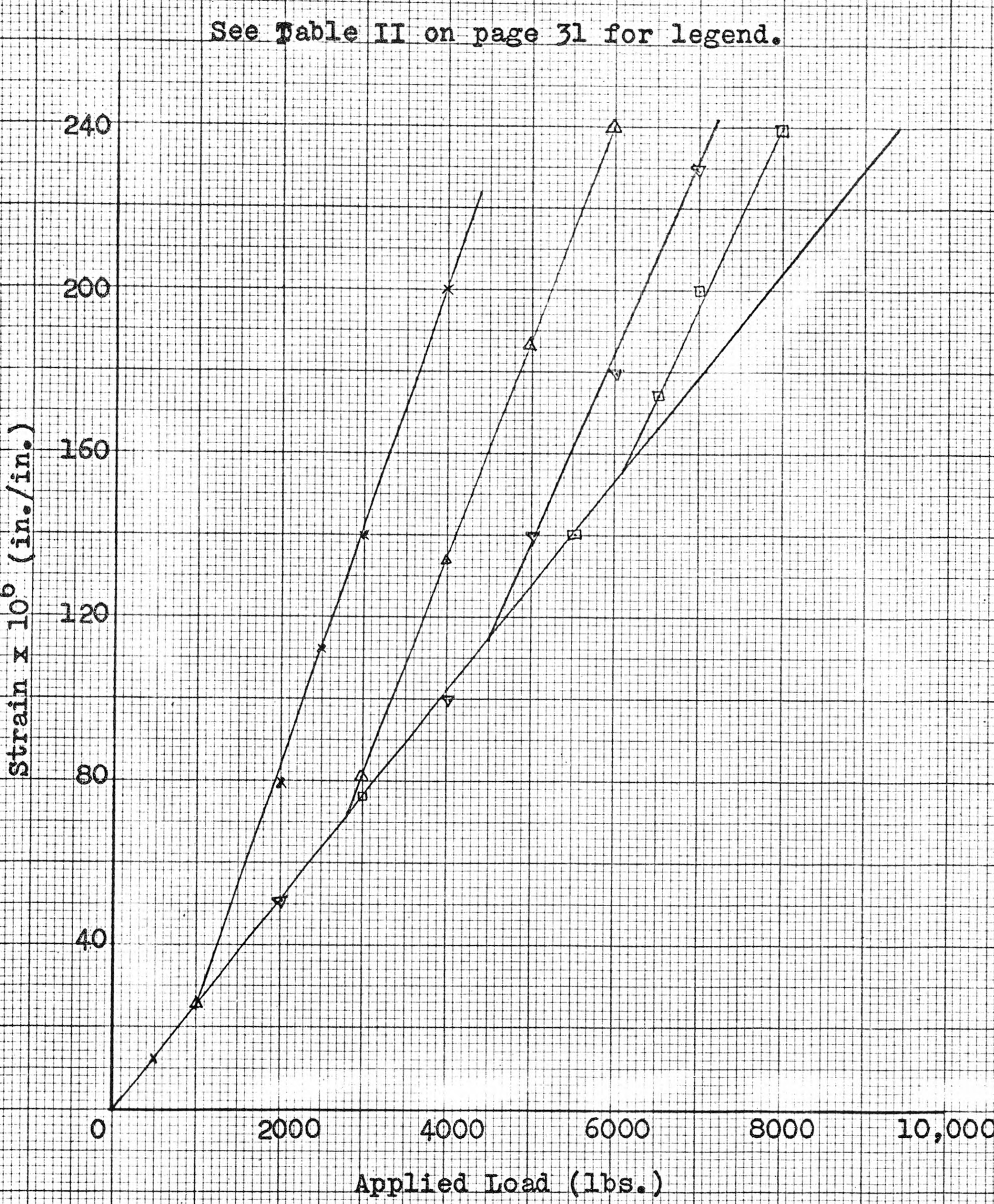

Figure 30. Tension Test with 3/8" Thick Copper Gasket

TABLE IX

PREDICTED AND ACTUAL VALUES OF LOAD CAUSING SEPARATION OF FLANGES FOR DIFFERENT GASKETS

Initial Torque (in. lb.)	Initial Strain $\times 10^6$ (in./in.)	Pretension (lbs.)	Load Causing Separation (lbs.) Calculated	Experimental
Rubber Gaskets:- 1/8" thickness				
150	151	889	1,175	900
300	398	2,342	3,100	3,100
450	665	3,920	5,175	4,900
600	885	5,525	6,900	7,300
Copper Gaskets:- 1/8" thickness				
150	144	849	1,120	1,000
300	403	2,375	3,130	2,800
450	674	3,980	5,250	5,250
600	954	5,620	6,050	6,500
Copper Gaskets:- 1/4" thickness				
150	164	966	1,245	1,350
300	396	2,339	3,000	3,000
450	724	4,265	5,490	5,800
600	956	5,640	7,260	7,850
Copper Gaskets:- 3/8" thickness				
150	128	754	910	1,000
300	350	2,061	2,490	2,800
450	633	3,730	4,500	4,500
600	910	5,360	6,475	6,100

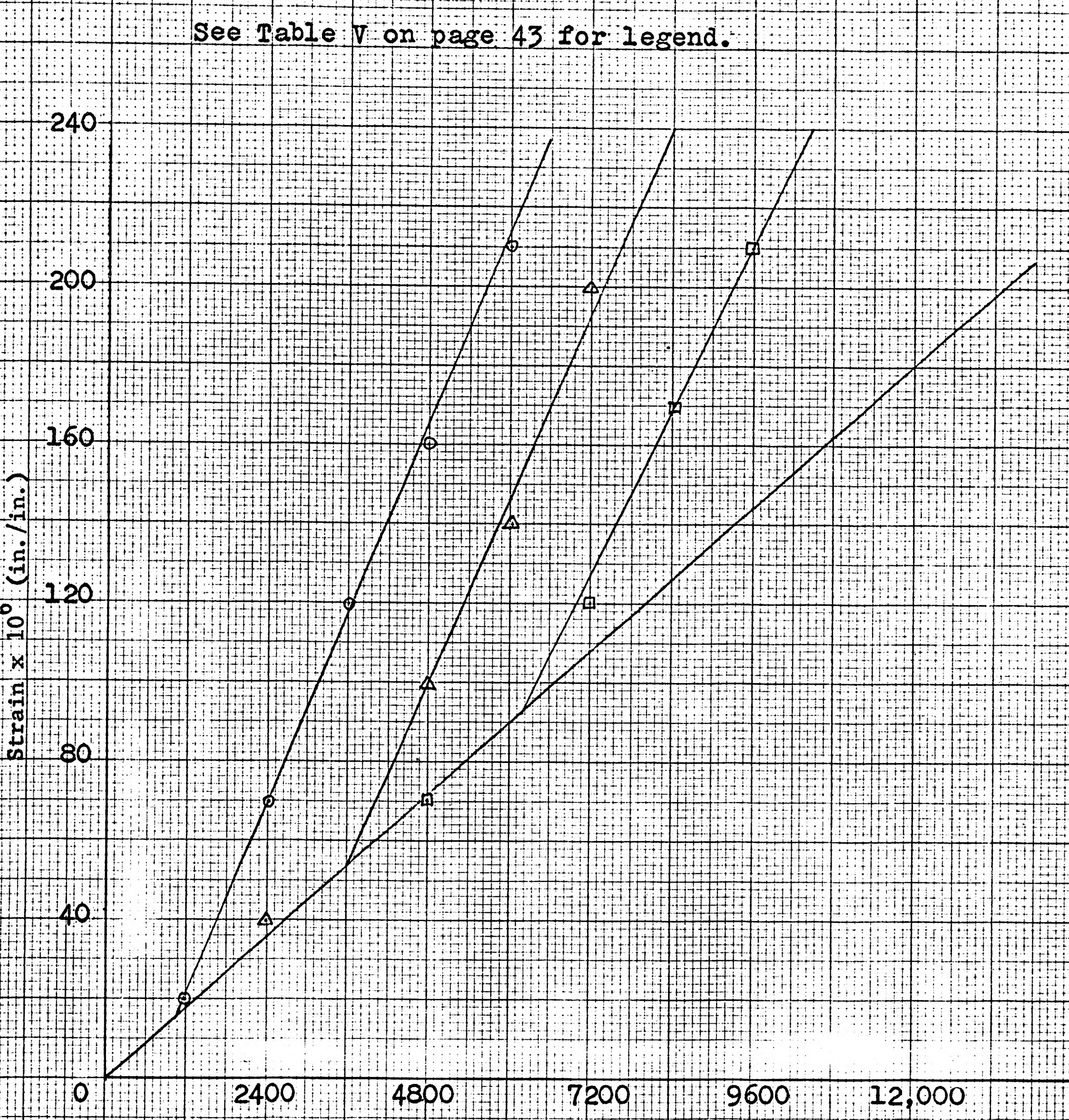

Figure 31. Bending Test with 1/8" Thick Rubber Gasket

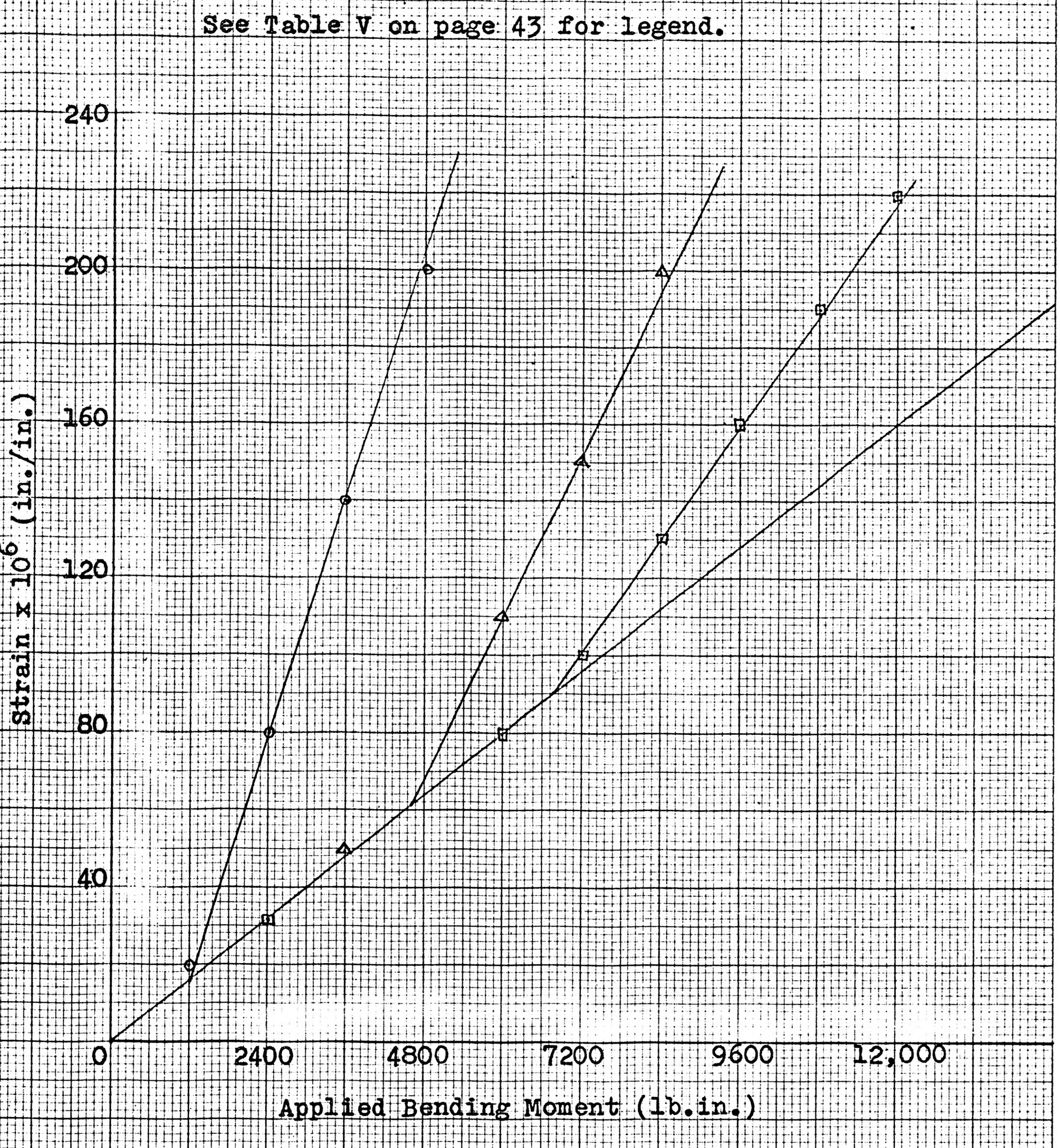

Figure 32. Bending Test with 1/8" Thick Copper Gaskets

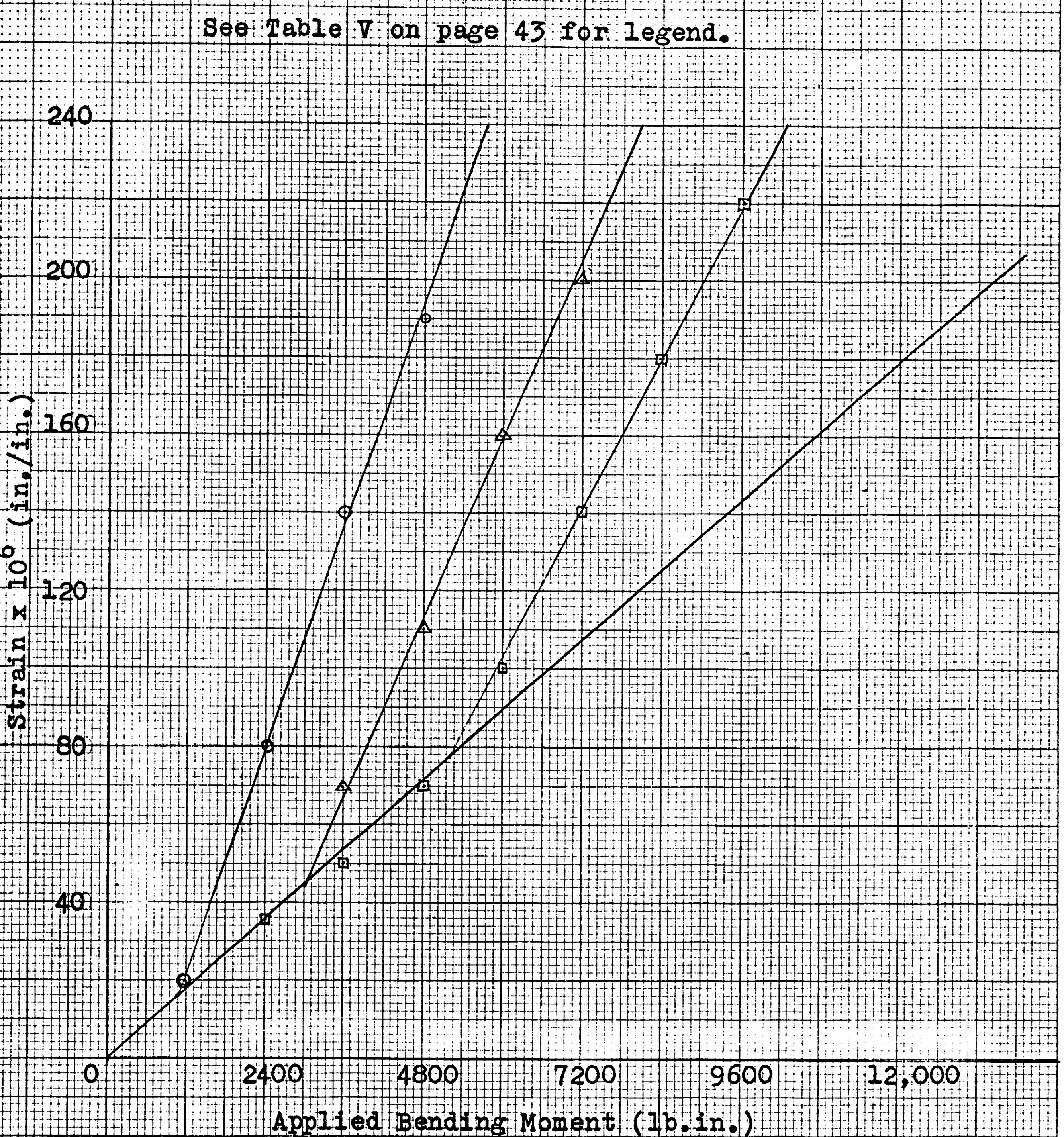

Figure 33. Bending Test with 1/4" Thick Copper Gasket

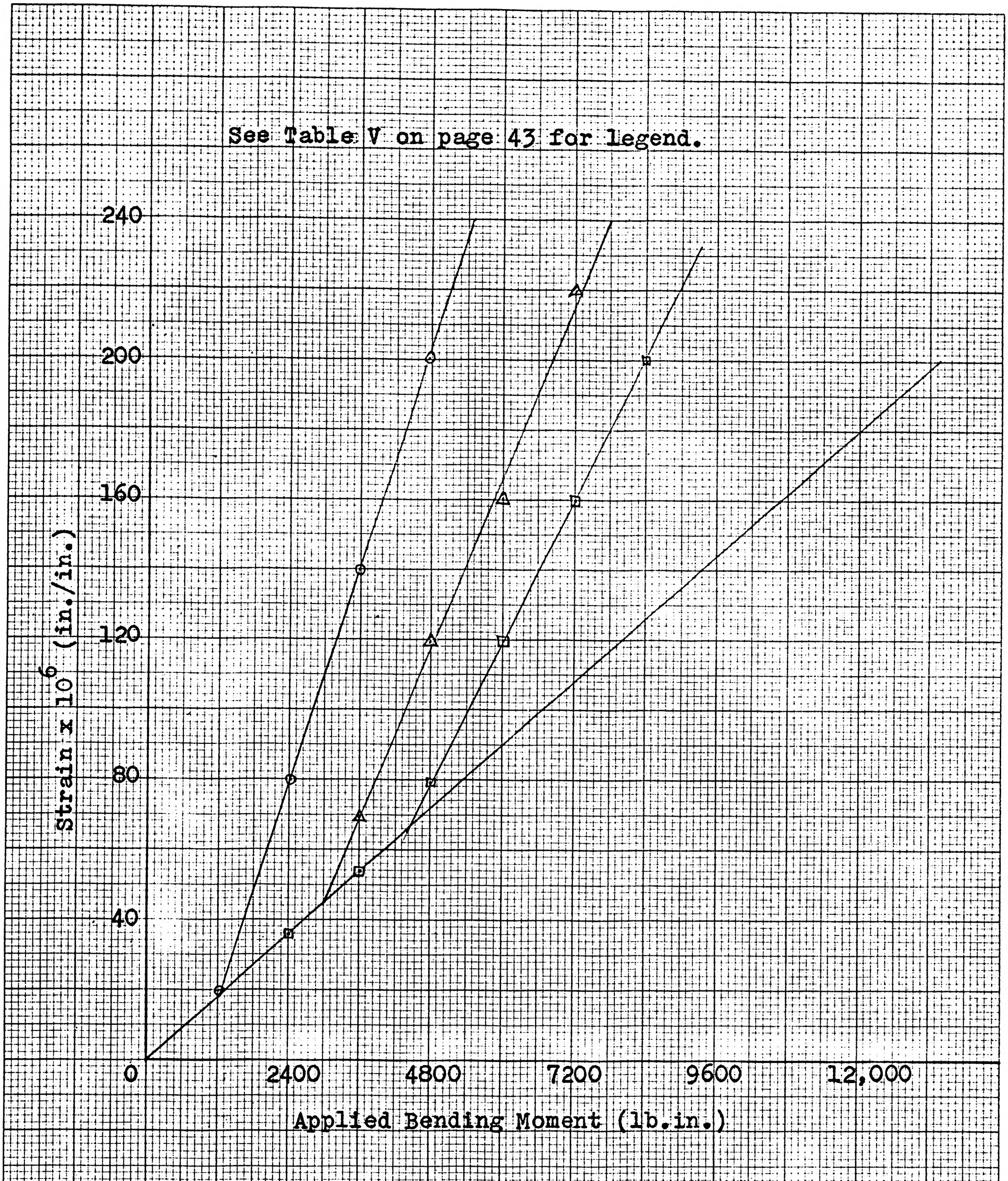

Figure 34. Bending Test with 3/8" Thick Copper Gasket

TABLE X

PREDICTED AND ACTUAL VALUES OF BENDING MOMENT CAUSING SEPARATION FOR DIFFERENT GASKETS

Initial Torque (in. lb.)	Initial Strain $\times 10^6$ (in./in.)	Pretension (lbs.)	Bending Moment Causing Separation (lbs.) Calculated	Experimental
Rubber Gaskets:- 1/8" thickness				
150	122	718	1,148	1,200
300	401	2,360	3,770	3,600
450	652	3,840	6,120	6,120
Copper Gaskets:- 1/8" thickness				
150	140	823	1,290	1,200
300	475	2,800	4,370	4,560
450	729	4,290	6,720	6,720
Copper Gaskets:- 1/4" thickness				
150	80	471	717	1,080
300	342	2,018	3,065	3,000
450	575	3,382	5,160	5,160
Copper Gaskets:- 3/8" thickness				
150	131	771	1,114	1,200
300	357	2,100	3,040	3,000
450	517	3,042	4,380	4,380

D. RESULTS OF TESTS:

Variation of K_1, K_2, and K_3 With Bolt Diameter:-

Bolt Diameter (in.)	Area of bolts (in.^{2})	K_1	K_2	K_3
3/8"	0.3318	0.0312	0.0674	0.7450
1/2"	0.5890	0.0405	0.0905	0.7460
5/8"	0.9210	0.0788	0.1448	0.7560
3/4"	1.3250	0.1200	0.1905	0.7480

Variation of K_1, K_2, and K_3 With Flange Diameter:-

Flange Diameter (in.)	Flange Area (in.^{2})	K_1	K_2	K_3
8.0"	50.30	0.0252	0.0579	0.7760
7.5"	44.20	0.0238	0.0729	0.7680
7.0"	38.50	0.0268	0.0771	0.7550
6.5"	33.20	0.0405	0.0905	0.7460
6.0"	28.30	0.0446	0.0975	0.6420

Variation of K_1, K_2, and K_3 With Gaskets:-

Type of Gasket	Thickness (in.)	K_1	K_2	K_3
Rubber	1/8"	0.0259	0.099	0.7360
Copper	1/8"	0.0279	0.1036	0.7530
Copper	1/4"	0.0241	0.0974	0.7720
Copper	3/8"	0.0208	0.0955	0.8150

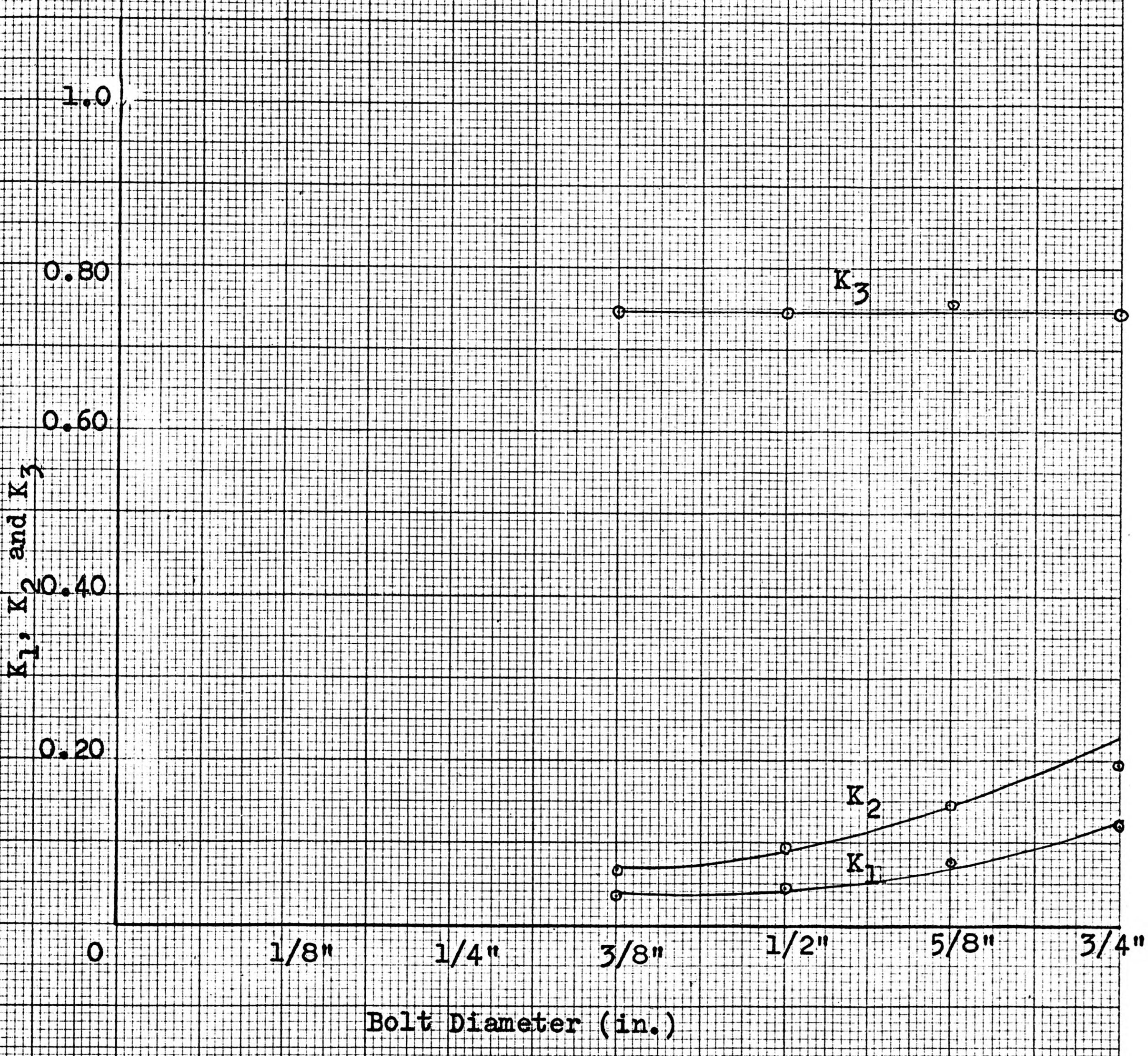

Figure 35. Variation of K_1, K_2 and K_3 with Bolt Diameter

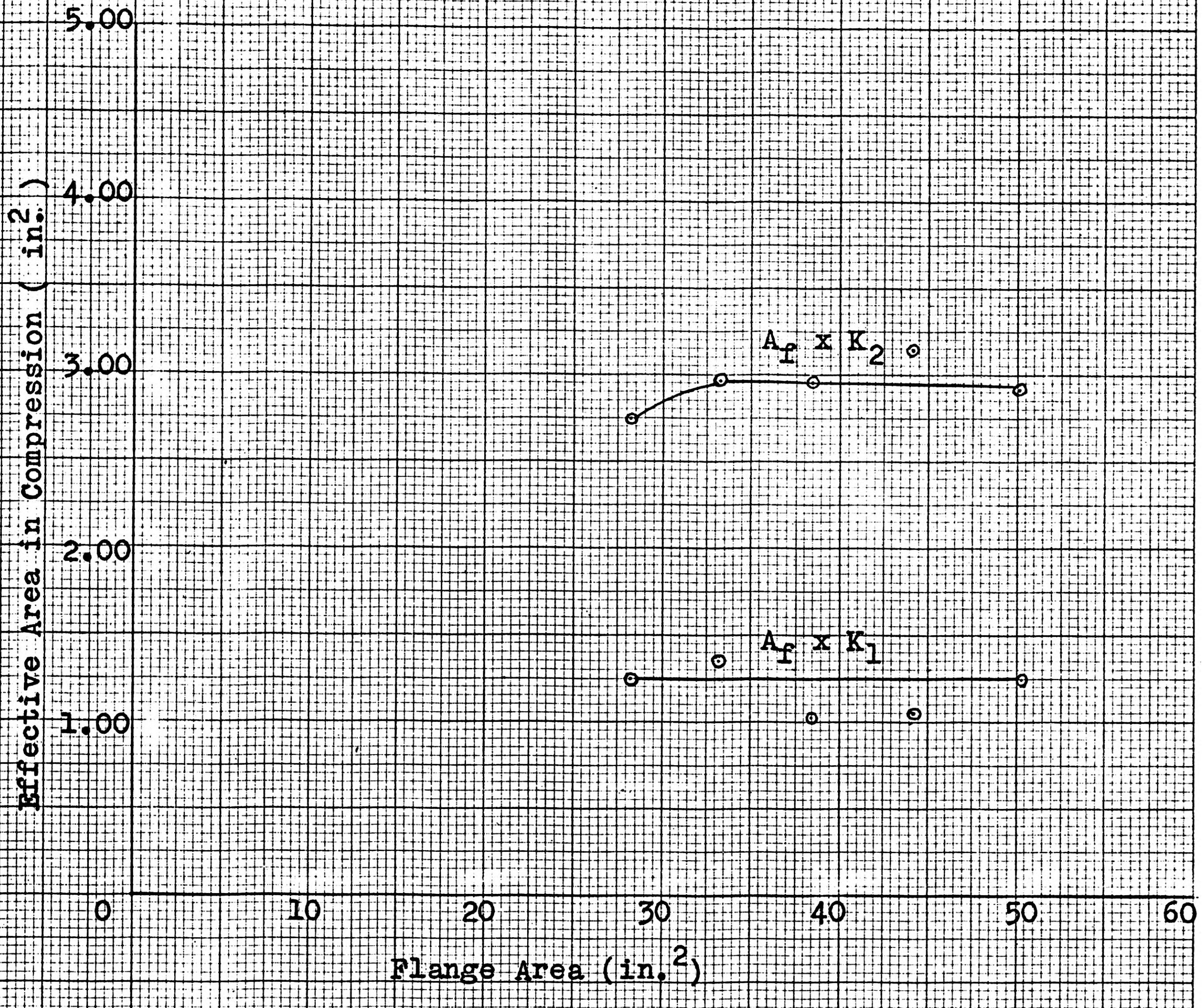

Figure 36. Variation of K_1 and K_2 with Flange Area.

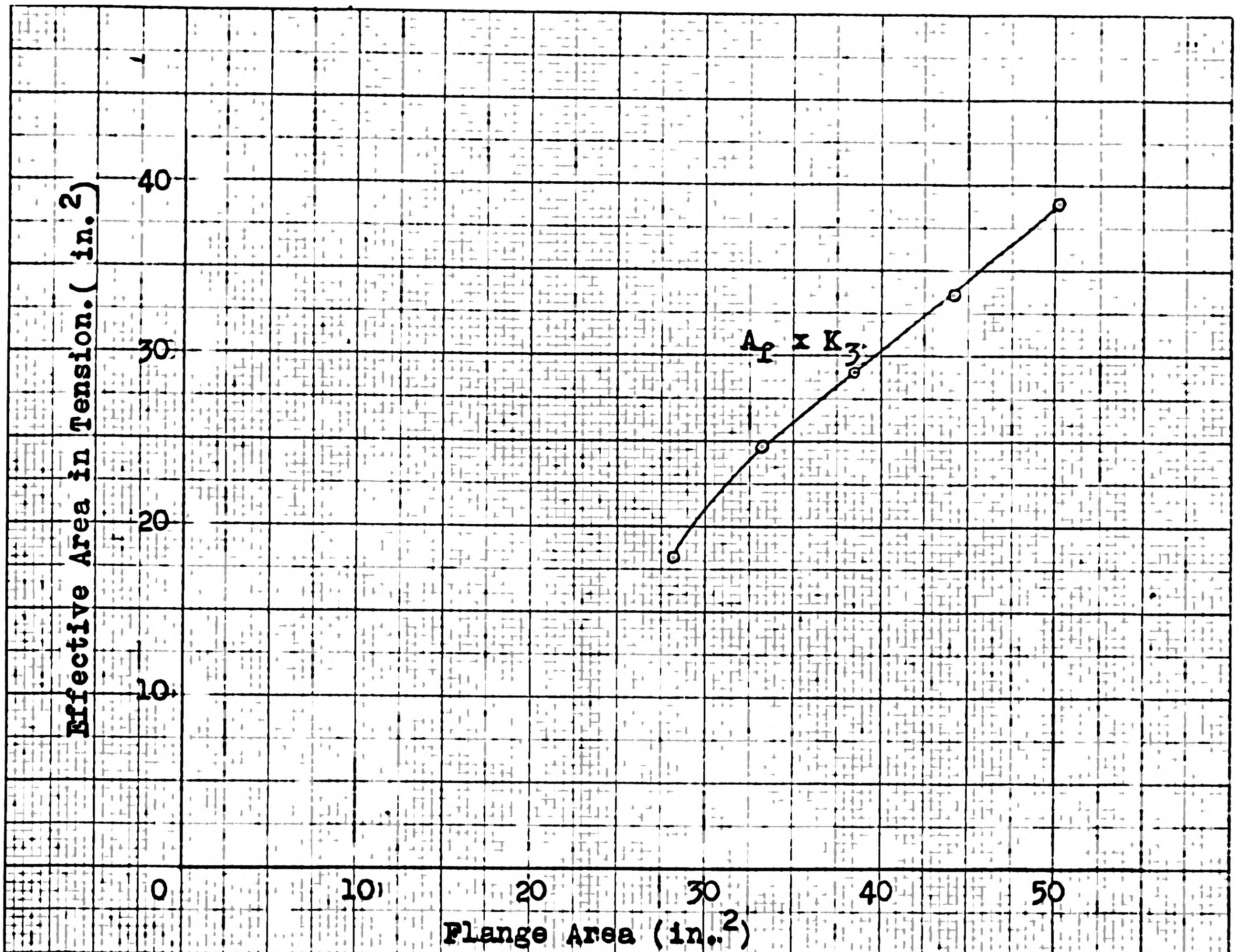

Figure 37. Variation of K_3 with Flange Area.

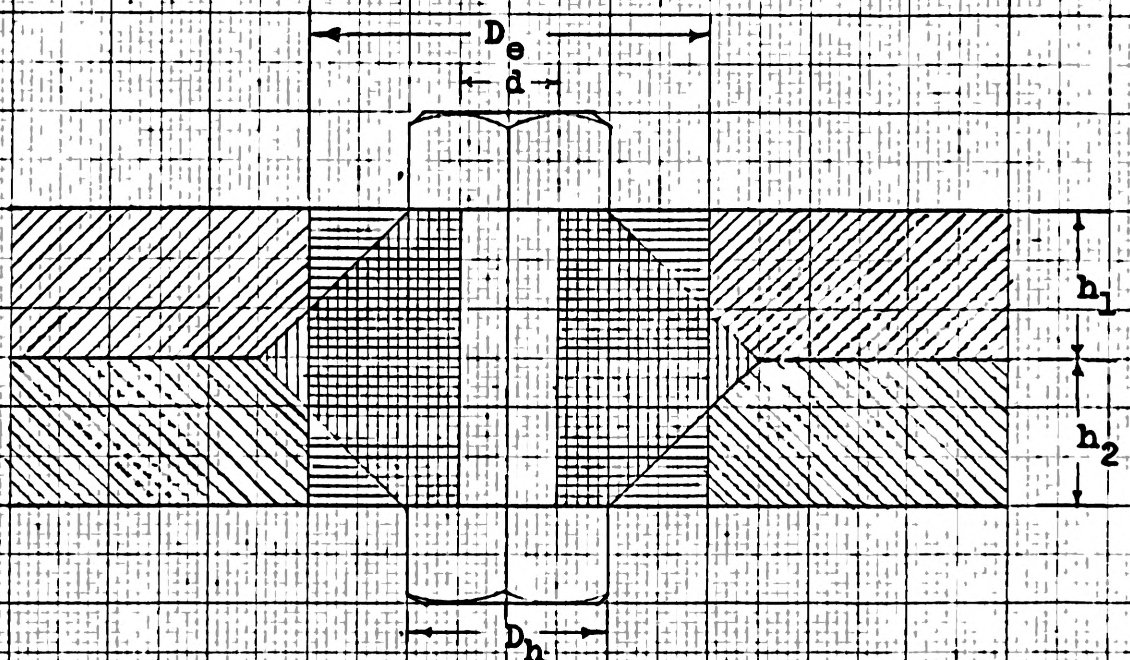

Figure 38. Schematic representation of Flanges undergoing Compressive deformation.

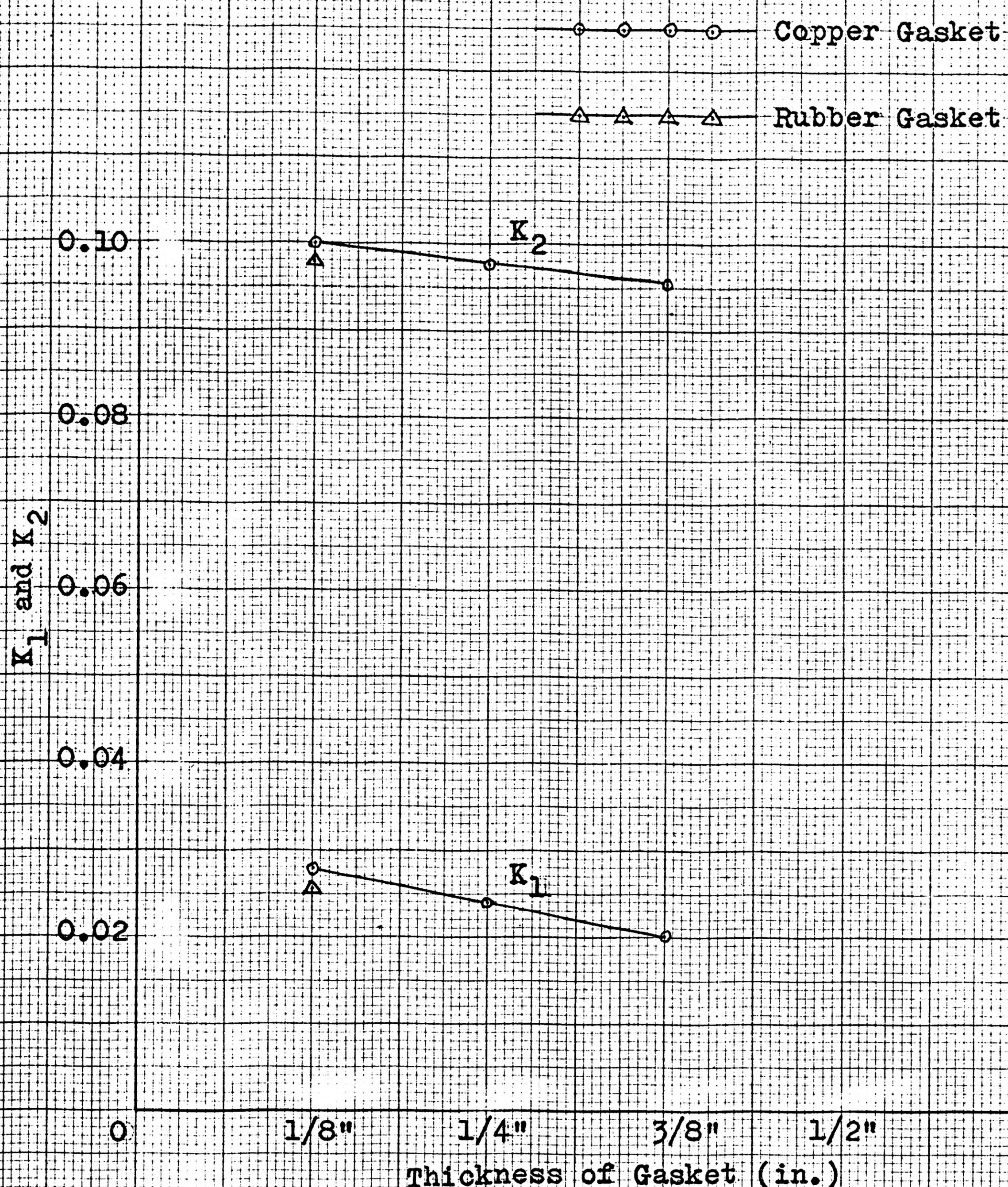

Figure 39. Variation of K_1 and K_2 with Gaskets

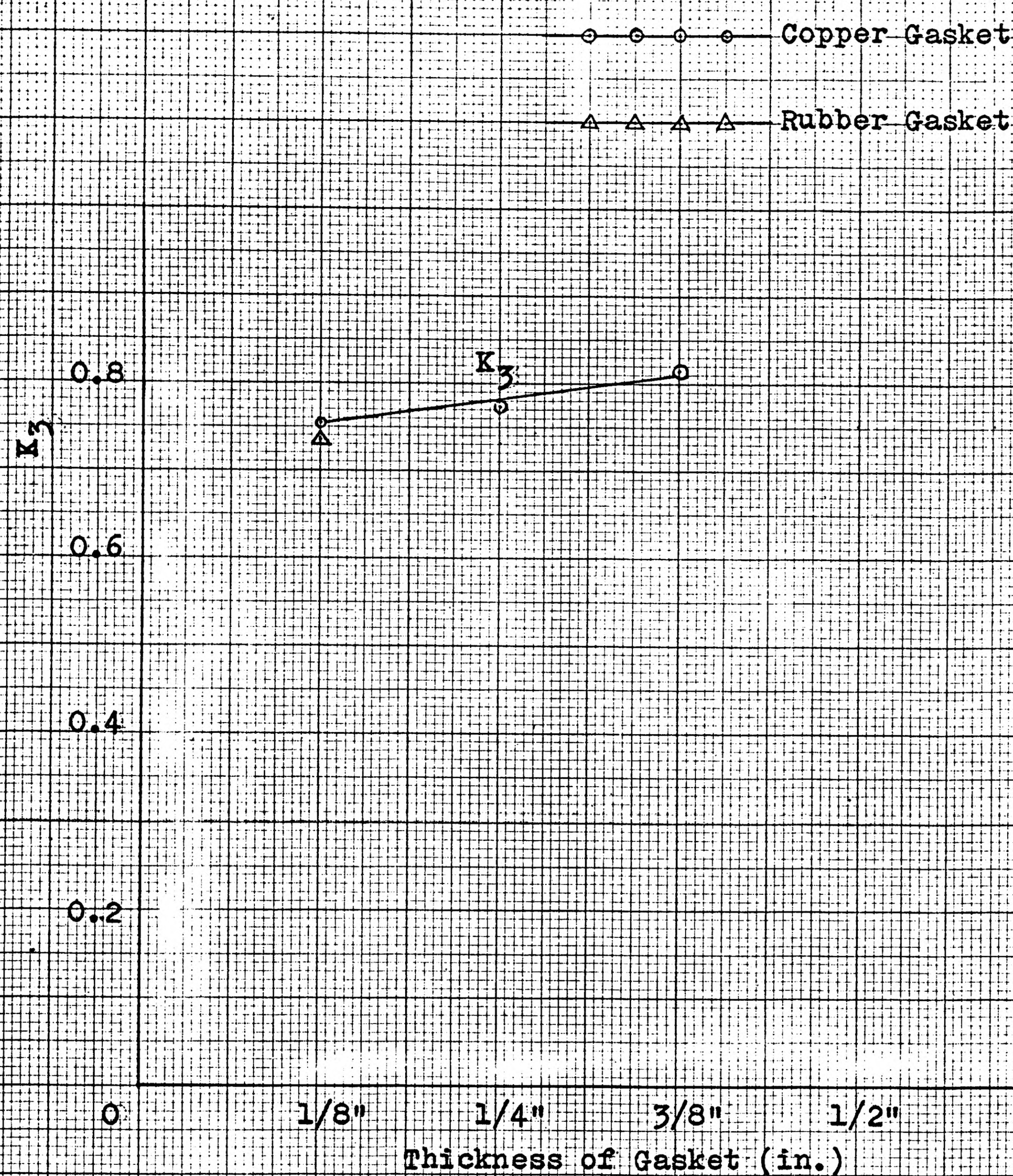

Figure 40. Variation of K_3 with Gaskets

V. CONCLUSION

The results of the experimental investigation are plotted on Figures 35-40. The following conclusions can be derived from these graphs and the graphs of flange coupling in tension and bending for various parameters.

A. Influence of Bolt Diameter on a Joint:

The variation of K_1, K_2, and K_3 with bolt diameter are plotted in Figure 35. The following points can be concluded from the experimental results.

(i) The stiffness of the joint in tension and bending increases as the bolt diameter increases.

(ii) The values of K_1 and K_2 increases as the bolt diameter increases. This indicates that the actual flange area in compression increases as bolt diameter. The compression of the flanges can be represented as shown in Figure 38.

Discussing the fatigue strength of a bolted joint Mr. E. I. Radzimovsky (2) states that the actual flange area in compression can be represented by compression cones as shown in Figure 38, which cut the bearing under the nut and the head of the bolt at an angle of about 45 degrees. Elasticity of a double cone can be determined approximately by replacing the cones with an equivalent hollow cylinder.

Thus, the effective cross-section area in compression is

$$A_c = \frac{\pi}{4} (D^2_e - d^2)$$

Where, D_e = Equivalent Diameter = $D_h + 1/2 (h_1 + h_2)$

D_h = Width across flat surfaces of the bolt

h_1 and h_2 = Thickness of the flanges.

d = Diameter of the bolt.

The results of the experiments were found to be fairly close to the values obtained by an equivalent hollow cylinder.

(iii) There is no significant effect of the bolt diameter on the bending moment causing separation of the flanges.

B. Influence of Flange Diameter on a Joint:

The variation of effective flange area in compression versus flange area is plotted in Figure 36. The following points can be concluded from the experimental results.

(i) The stiffness of the joint in tension does not necessarily increase as the flange diameter. This is apparently clear from Figure 38, which shows the actual flange area under compression. Hence, the increase in flange diameter beyond standard size does not increase the compression area of the flange.

(ii) The stiffness of the joint in bending increases as the flange diameter increases. The bending moment causing separation of the flanges increases as the diameter of the flange.

C. Influence of Gaskets on a Joint:

The variation of K_1, K_2 and K_3 with type of gaskets and thickness of the gaskets are plotted in Figure 39 and 40. It can be concluded that the metallic or non-metallic gaskets reduce the stiffness of the joint. Moreover, increase in thickness of the gaskets causes considerable reduction in the strength of the joint. It should be noted that the accuracy of the equipment available was limited and so the results are conclusive to a limited extent.

The Technical Information Staff of Industrial Fasteners Institute (5) reports that the use of a gasket in a joint increases the flexibility of a joint and increases the external load felt by a bolt. The effect of external forces increases with a softer gasket. Various other methods like solder plug, etc. have been used in the past to determine gasket loads which are more accurate and conclusive than the method used here. (6)

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VITA

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